OL. XX — N° 1 JANUARY 1938

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MONTHLY BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS ASSOCIATION

[385. (05]

VOL. XX. — 1938.

NOTICE TO READERS

LIST OF ABBREVIATIONS OF MEASURES, ETC., USED IN THE *BULLETIN*

Kilometre	km.	Hour	h.
Metre	m.	Minute	m.
Centimetre	cm.	Second	s.
Square kilometre	km^2.	Kilogrammetre	kgrm.
Square metre	m^2.	Kilowatt	kw.
Square centimetre	cm^2.	Kilowatt-hour	kw.-h.
Hectare	ha.	Watt	w.
Are	a.	Horse power	H. P.
Centiare	ca.	Per cent	%
Cubic metre	m^3.	Per thousand	‰
Cubic centimetre	cm^3.	Tonne-kilometre	tkm.
Stere	st.	Passenger-kilometre	p.-km.
Hectolitre	hl.	Train-kilometre	tr.-km.
Litre	l.	Franc	fr.
Centilitre	cl.	Centime	c.
Degree	°	Lire	L.
Minute	′	Centesimo	c.
Second	″	Pound sterling	£.
Tonne (Metric)	t.	Shilling	sh.
Ton (English).	ton.	Penny	d.
Hundredweight	cwt.	Dollar	$.
Kilogramme	kgr.	Cent	ct.
Gramme	gr.	Reichsmark.	Rm.
Centigramme	cgr.	Pfennig	pf.

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MONTHLY BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS ASSOCIATION

(ENGLISH EDITION)

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Vol. XX. — No. 1. January 1938.

BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[621. 132.3 (.436) & 621. 132.8 (.436)]

New steam locomotives of the Austrian Federal Railways.

by EMMERICH KARNER,
Engineer, Superintendent of Motive Power, Austrian Federal Railways.

The European economic crisis has naturally also had its effects on the Austrian Federal Railways. Both the passenger traffic and the goods traffic have decreased very considerably. The repercussions of the crisis were still further aggravated by motor-car and motor-bus competition, a competition which was particularly severe as regards the conveyance of passengers, so that the Administration was very quickly obliged to make an appreciable reduction in its services, either by running, on certain lines, smaller transport units at more frequent intervals than the usual local trains, and consequently able to give the passengers the advantages of a more frequent service between their places of residence and the economic centres.

A service of this kind might also have been met by railcars, but only for lighter loads, because for reasons of economy, the power of the internal combustion engines employed, and consequently the first costs, could not be very high. It will readily be understood that such motor vehicles are unsuitable for very variable train loads, their reserve of power, limited on the score of price, being too low.

The problem thus raised consisted of the construction of a vehicle, rather low in first cost, economical in service, and capable of meeting large fluctuations in the loads. Other conditions to be fulfilled were high speed, and a weight per axle as low as possible to enable the vehicle to be run on a light track, and lastly the possibility of the vehicle being driven by one man only. In order to fulfil all these conditions, it appeared necessary to build a steam locomotive.

With a view to securing the greatest possible safety in service, it was decided to use no type of boiler or motion which could not be made by the manufacturing and machining methods already used in railway service. This led to the adoption of a normal locomotive boiler and the motion of a steam locomotive, which were known to be absolutely reliable in operation.

The question which then had still to

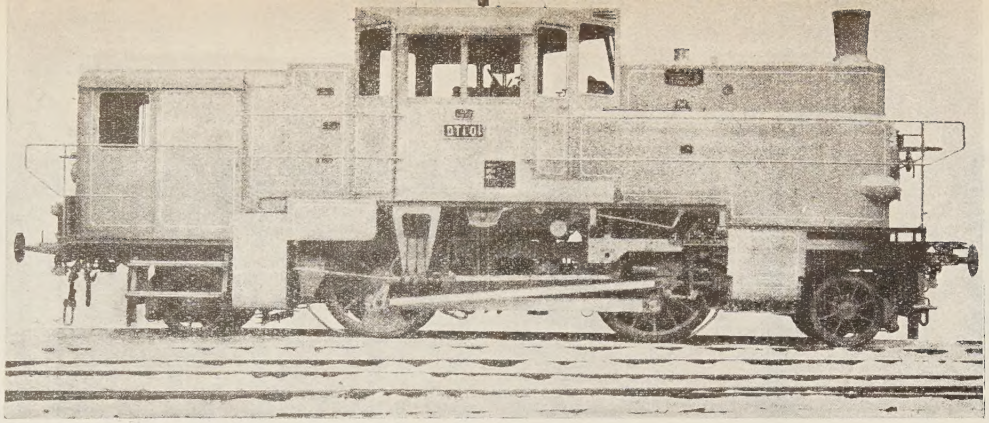
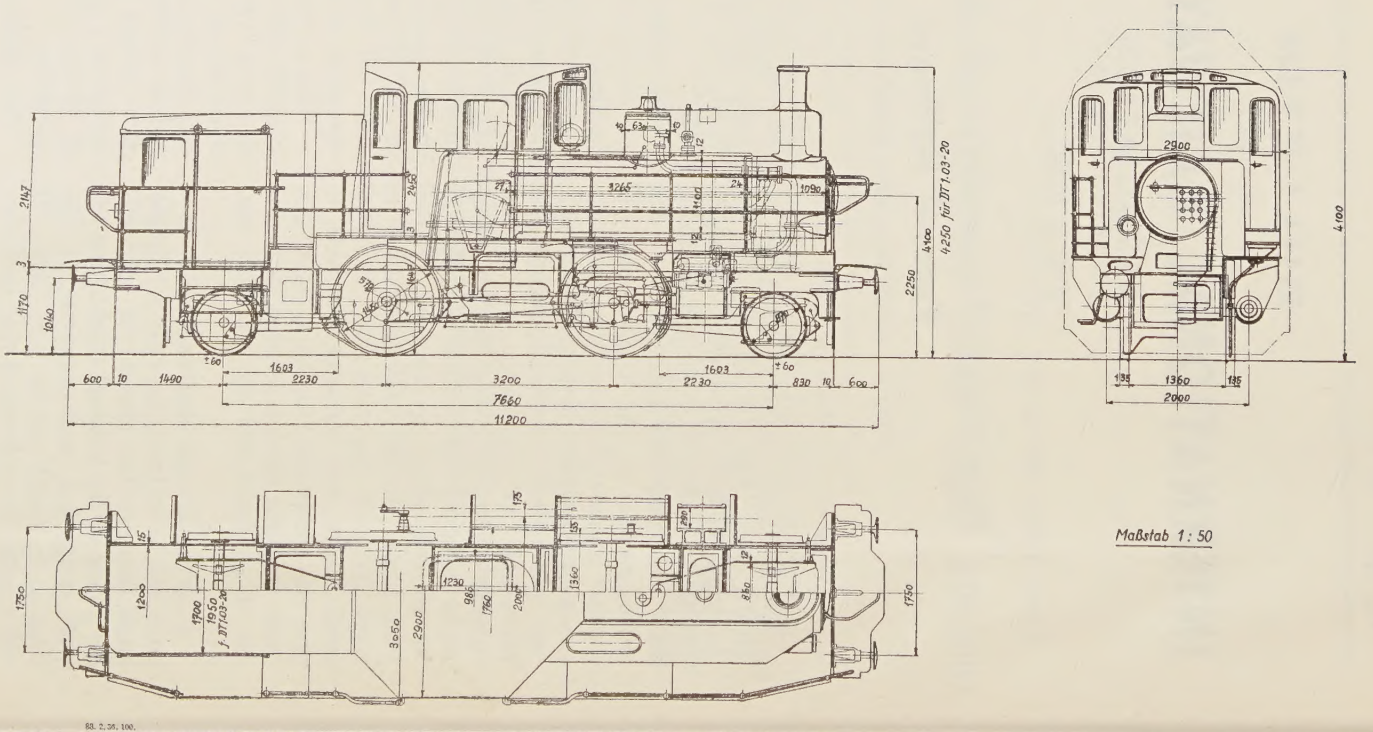


Fig. 1. — Steam rail motor van.



Grate area	0.83 m² (8,93 sq. ft.)	0.83 m² (8,93 sq. ft.)
Ordinary tubes — Number	35	35
Ordinary tubes — Diameter (outside/inside)	41.5/37 mm. (1 5/8" — 1 29/64")	41.5/37 mm. (1 5/8" — 1 29/46")
Large tubes — Number	24	24
Large tubes — Diameter (outside/inside)	108/100.5 mm. (4 15/64" — 3 15/16	108/100.5 mm. (4 15/64" — 3 15/16")
Heating surface, fire side/water side — Firebox	4.45/4.50 m² (47.9/48.4 sq. ft.)	4.45/4.50 m² (47.9/48.4 sq. ft.)
Heating surface, fire side/water side — Ordinary tubes	13.30/14.90 m² (143.2/160.4 sq. ft.)	13.30/14.90 m² (143.2/160.4 sq. ft.)
Heating surface, fire side/water side — Large tubes	24.73/26.60 m² (266.2/286.3 sq. ft.)	24.73/26.6 m² (266.2/286.3 sq. ft.)
Evaporating heating surface — water side	46.0 m² (495 sq. ft.)	46.0 m² (495 sq. ft.)
Evaporating heating surface — fire side	42.48 m² (457.3 sq. ft.)	42.48 m² (457.3 sq. ft.)
Superheating surface — fire side	21.40 m² (230.3 sq. ft.)	21.40 m² (230.3 sq. ft)
Superheating surface — steam side	16.47 m² (177.3 sq. ft.)	16.47 m² (177.3 sq. ft.)
Safety valves, Coale type, 2". Number	2	2
Effective steam pressure	16 atm. (227.6 lb. per sq. in.)	16 atm. (227.6 lb. per sq. in.)
Diameter of driving wheels on tread, tyres 50 mm. (2") thick	1 410 mm. (4' 7 1/2")	1 410 mm. (4' 7 1/2")
Diameter of carrying wheels on tread, tyres 50 mm. (2") thick	830 mm. (2' 8 5/8")	830 mm. (2' 8 5/8")
Driving wheels — Diam. of journals	160 mm. (6 5/16")	160 mm. (6 5/16")
Driving wheels — Length of journals	205 mm. (8 1/4")	205 mm. (8 1/4")
Driving wheels — Distance centre to centre of journals	1 120 mm. (4' 8")	1 120 mm. (4' 8")
Coupled wheels — Diam. of journals	160 mm. (6 5/16")	160 mm. (6 5/16")
Coupled wheels — Length of journals	205 mm. (8 1/4")	205 mm. (8 1/4")
Coupled wheels — Distance centre to centre of journals	1 120 mm. (4' 8")	1 120 mm. (4' 8")
Carrying wheels — Diam. of journals	130 mm. (5 1/8")	130 mm. (5 1/8")
Carrying wheels — Length of journals	255 mm. (8' 4 25/64")	255 mm. (8' 4 25/64")
Carrying wheels — Distance centre to centre of journals	1 130 mm. (3' 8 1/2")	1 130 mm. (3' 8 1/2")
Weight in working order, per linear metre (*per linear foot*), tyres 50 mm. (2") thick, reckoned on the length over the buffers	3.78 t (Engl. tons)	4.06 t. (Engl. tons)
Diameter of cylinders	290 mm. (11 7/16")	290 mm. (11 7/16")
Piston stroke	570 mm. (22 7/16")	570 mm. (22 7/16")
Length of connecting rods	3 350 mm. (11')	3 350 mm. (11')
Outside valve gear, type: Walschaerts.		
Inside valve gear, type: Lentz, with oscillating cams.		
Capacity of the water tank (3.5 m³ for 42 t. of loaded weight)	4.70 m³ (1 034 Br. gall.)	5.30 m³ (1 166 Br. gall.)
Capacity of the coal bunker (1.1 t. for 42 t. of loaded weight)	1.76 t. (1.73 Engl. tons)	1.76 t. (1.73 Engl. tons)
Capacity of the oil reservoir (0.5 t. for 42 t. of loaded weight)	1.17 t. (1.15 Engl. tons)	1.17 t. (1.15 Engl. tons)
Area of floor space in luggage compartment	4.40 m² (47.3 sq. ft.)	5.00 m² (53.8 sq. ft.)
Luggage	1 000 kgr. (2 200 lb.)	1 000 kgr. (2 200 lb.)
Empty weight, with tyres 50 mm. (2") thick	34.2 t. (33.7 Engl. tons)	34.2 t. (33.7 Engl. tons)
Weight fully loaded, with tyres 50 mm. (2") thick — 1st pair of wheels	9.9 t. (9.74 Engl. tons)	10.18 t. (10.02 Engl. tons)
Weight fully loaded, with tyres 50 mm. (2") thick — 2nd pair of wheels	13.0 t. (12.79 Engl. tons)	13.0 t (12.79 Engl. tons)
Weight fully loaded, with tyres 50 mm. (2") thick — 3rd pair of wheels	13.0 t. (12.79 Engl. tons)	13.0 t (12.79 Engl. tons)
Weight fully loaded, with tyres 50 mm. (2") thick — 4th pair of wheels	8.9 t. (8.76 Engl. tons)	9.37 t. (9.22 Engl. tons)
	44.8 t. (44.08 Engl. tons)	45.55 t. (44.82 Engl. tons)
Maximum speed	100 km./h. (62 m. p. h.)	100 km./h. (62 m. p. h.)
Minimum radius of curves	150 m. (7 1/2 chains)	150 m. (7 1/2 chains)

Brake: Type T vacuum brake of locomotive and vehicles.

Amplification $\ddot{u} = \frac{324}{115}\left(\frac{120}{120}\cdot\frac{240}{120}+\frac{240}{120}\cdot\frac{180}{180}\right) = 11.27$ for the coupled axles.

$\ddot{u} = \frac{300}{80}\cdot\frac{415}{205} = 7.59$ for the bissel truck.

Pressure of the brake blocks:

$(\eta = 0.95)\ \frac{0.95 \cdot 4.80 \cdot 11.27}{4} = 4.85$ t. (4.77 E gl. tons) per wheel.

$\frac{0.95 \cdot 700 \cdot 7.54}{2} = 2.52$ t. (2.48 Engl. tons) per wheel.

Ratio of braking to total weight — Coupled wheels	78.5 %	78.5 %
Ratio of braking to total weight — Bissel truck	66.5 %	65 %
Ratio of braking to total weight — Coupled wheels and carrying wheels	73.8 %	68.5 %

Hand brake:

Amplification $\ddot{u} = \frac{3\,\pi}{25.4\ \cdot\ \frac{2}{5}}\cdot\frac{185}{200}\cdot\frac{190}{115}\cdot 4 = 718.3$

Pressure of brake blocks

$(\eta = 0.3)\ \frac{718.3 \cdot 50 \cdot 2.3}{4} = 2.69$ t. (2.65 Engl. tons) per wheel.

Ratio of braking to total weight	25.6 %	23.7 %

be solved was the arrangement of the wheels. For hauling light trains a « B » (0-4-0) locomotive would have sufficed, but it would not have had the running qualities necessary for the high speeds — up to 100 km. (62 miles) per hour — which were required. The 1-B (2-4-0) type locomotive did not meet the case either, because the engine had to be capable of running with equal facility in either direction. A 1-B-1 (2-4-2) engine was therefore built, with leading and trailing bissel trucks (see figs. 1 and 2).

As the tractive effort aimed at naturally implied a boiler of small dimensions, with, however, a considerable length between the supports on account of the high speeds, advantage was taken of the extremely long frame to add to the driver's cab, in addition to the coal bunker, a luggage compartment with a floor area of 4.40 m² (47.4 sq. ft.), capable of holding about 1 ton of luggage, so that luggage vans were not required on the short trains. However, a place had to be reserved on the locomotive for the guard, and a safe communication arranged between the driver's cab and the coaches, which was carried out by converting the footplate into side corridors.

As stated above, the locomotive was fitted with a boiler of the normal type. For the first time in Austria, the copper firebox was welded. Because of the small dimensions — the grate area is only 0.83 m² (8.93 sq. ft.) — it was difficult to weld the joints, the welder being hampered in his movements inside the firebox. For the engines which were ordered subsequently, the welding of the fireboxes was abandoned in favour of riveted fireboxes. On both sides of the rectangular fire door which opens inside, holes are arranged for the oil burners. The boiler barrel is formed of a single plate of high-grade steel. In front of the boiler barrel is the smokebox which, in the case of this engine, is fitted with a large pocket for the cinders, accessible from the platform through a cleaning door. The bottom of this pocket is flat and covered with a layer of cement.

As it is necessary to start the locomotive very frequently, and the distances between stations are short, care was taken to fit a superheater of such dimensions as would enable high temperatures to be obtained quickly . The superheater is of a completely new type. In order to save weight, the saturated and superheated steam headers were welded together with steel tubes of square section. As these two headers were placed one behind the other in the smokebox, it was necessary for the superheater elements to be arranged in such a manner that their ends should not be connected up to the headers, side by side, but one behind the other. For this reason, the method of fixing the tube ends in a common wrought iron header had to be abandoned. The flanges of the main steam pipes are welded to the front wall of the superheated steam header. Similarly, the tubes which connect the saturated steam header with the boiler are electrically welded to the header. The cylinder inlet and exhaust pipes are arranged in such a way as to afford them free expansion, The exhaust pipes discharge into a welded breeches pipe which is connected with the blast pipe column.

The regulator, which is of the Zara type, is operated in the usual manner by side rods.

The two cylinders of this locomotive are perfectly identical and interchangeable. They were both cast integral with their steam chests. In Austria, the steam chests are generally cast separately and

assembled with the cylinders by means of flanges. On account of the smallness of the cylinders of the engine in question, the pipes leading from the steam chests to the cylinders would have been too long with the assembled steam chests, and consequently the clearance would have become too great. For this reason, the standard type, which is so convenient as regards renewals and repairs, was abandoned in favour of the construction of the cylinder and the steam chests in one piece. The poppet valves, made of wrought iron, are moved by a valve motion of the Lentz type, with adjustable counter-links. Whilst the engine is running with the regulator closed, the bye-pass valves, actuated by a small steam piston, lift all the cylinder poppet valves off their seats, and the result of this is that the pressure becomes perfectly balanced. The springs controlling the poppet valves are placed on the atmosphere side and are adjustable.

The outside valve motion is of the Walschaerts type which, like the regulator, can be operated from both sides of the driver's cab by means of levers. These levers mesh into two toothed segments, which can simultaneously be thrown in or out of gear by pedals worked by the driver.

The piston heads are of the hollow type made of cast iron. The piston rods are attached by a conical head in the crossheads running on a single slidebar. As in the case of the large 214 class express locomotives of the 2-8-4 type, the crossheads have been provided with gudgeon pins having spherical bearings for the purpose of eliminating the torsion stresses exerted on the connecting rods. It was necessary to make the coupling rod of exceptionally heavy section on account of its extreme length (3.20 m. = 10 ft. 6 in.) and the high running speeds required. Like the connecting rods, the coupling rods are fitted with solid ends. In order to permit easy dismantling of the rods from above the driving crank-

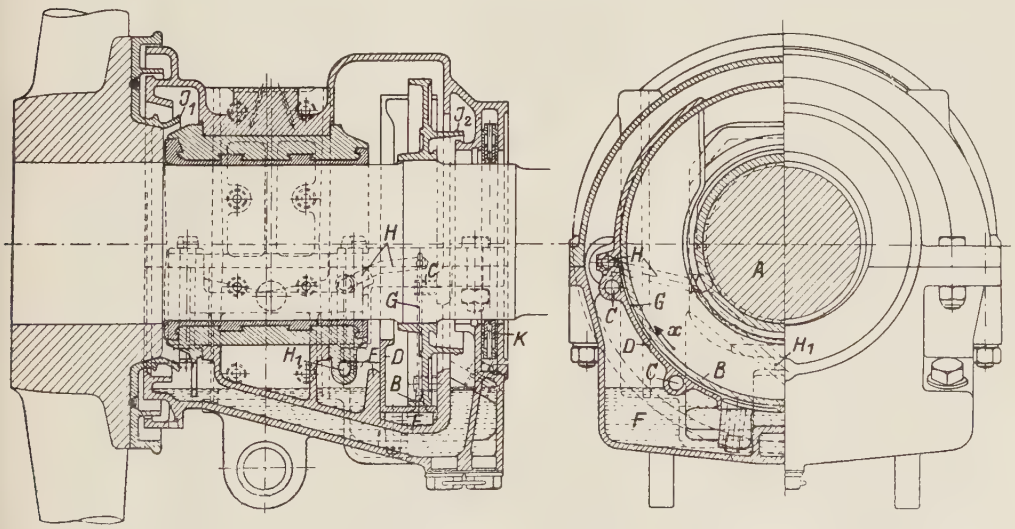


Fig. 3. — Inside axlebox, Friedmann type.

pins, the return cranks are pressed on these crankpins and fixed by means of set-screws.

Bearings for the driving, coupled and carrying wheel axles lubricated by a forced-feed circulating system of the Friedmann type have been used for the first time on the Austrian Federal Railways on these locomotives. Figure 3 shews the arrangement of a bearing of this type. An oiling drum B is shrunk and keyed on the axle A. This drum revolves in the fixed casing D which the springs and caps C keep pressed against the axlebox E. When the axle revolves in the direction of the arrow X, the drum B forces the oil from the well F into the groove G in the casing D. On leaving through the end of this groove, the oil is forced through the pipes H on to the rubbing surfaces. The superfluous oil is led through the pipe H_1 on to the other side of the bearing, whence it returns to the lubricating device. If the journal revolves in the opposite direction, the oil delivery also changes direction. For the purpose of avoiding leakages of oil the splash rings J_1 and J_2 have been fitted (revolving with the axle) at the points where the axle projects from the box. Oil wastage is prevented by these rings, in conjunction with the box itself, which is of a suitable shape. Protection against dust and water getting in is ensured by a dust shield in several pieces fitted towards the middle of the axle in a pocket in the box; a baffle formed of the splash ring and the box body serve the same purpose. The provision of these bearings on the small locomotives, the wheels of which are only 1.450 m. (4 ft. 9 in.) in diameter, has enabled speeds of 110 to 120 km. (68 to 75 miles) per hour to be reached and maintained.

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the end of a certain distance, and which, in conjunction with the automatic oil firing, enables the engines to be driven by one man only. Like all the other modern locomotives of the Austrian Federal Railways, the locomotive described in the present article is fitted with an exhaust steam pre-heater of the Heinl type, the thermal storage reservoir for which is housed in the lateral water tank on the left side. Besides the two tanks arranged alongside the boiler, a large water tank is also fitted under the coal bunker, right across the locomotive, behind which is placed, also across the frame, the liquid fuel tank.

The driver's cab is situated sufficiently high to give the driver a clear view of the track in either running direction. It adjoins, in the rear, the coal bunker and the luggage compartment and is accessible from the front and rear platforms through four doors. All the fittings and mountings are arranged in the cab in such a manner as to be capable of operation from the fireman's side as well as from the driver's side. This is particularly the case with the reversing gear, the lever for operating the regulator, and the control of the oil firing.

A 500-watt turbo-dynamo ensures the supply of current for lighting, as well as the current required for the head lights fitted at both ends of the locomotive. The coaches which are equipped for electric lighting can also be connected up to this lighting installation.

The small locomotive described above, which has received the designation of « steam rail motor van Class DT.1 » very rapidly proved its worth in service, and was employed on a wide variety of jobs. It was soon used, for example, to haul short fast trains — weighing about 80 t. (78.7 Engl. tons) — on the Semmering line whereon there are 1 in 40 gradients, up which it hauled this load at a speed of 28 km. (17.4 miles) an hour. The indicated horse-power developed on the trial run was as high as 450, with a coal consumption of 1 000 kgr. per m^2 (204 lb. per sq. ft.) of grate area per hour. A particularly noteworthy fact was the high degree of superheat obtained, even for low consumption figures, and this was undoubtedly one of the principal reasons for the extremely economical results given by these locomotives. Under heavier conditions of working, degrees of superheat exceeding 450° C. (842° F.) were obtained.

The great increase in traffic during 1926 and 1927 on the main lines of the Austrian Federal Railways resulted in the 2-D (4-8-0) and 1-C-2 (2-6-4) locomotives of the 113 and 310 classes, exclusively used hitherto, being unable to cope with the express train services. The Federal Railways therefore were compelled to design new and more powerful locomotives. The first point to be settled was the number of coupled pairs of wheels. It had, moreover, to be decided whether the desired reduction in the journey time was to be obtained by greatly increasing the running speed or by rapid acceleration after stopping or slowing down. In the first case, a heavy locomotive with large diameter wheels and three coupled pairs of wheels would be necessary; while in the second a locomotive with wheels of medium diameter and four coupled pairs of wheels would be better. Two locomotives were designed and their timings calculated for the Vienna-Salzburg run.

This showed that the four-coupled engine with rapid acceleration was very much better than the three-coupled, suitable for hauling the trains at high

pins, the return cranks are pressed on these crankpins and fixed by means of set-screws.

Bearings for the driving, coupled and carrying wheel axles lubricated by a forced-feed circulating system of the Friedmann type have been used for the first time on the Austrian Federal Railways on these locomotives. Figure 3 shews the arrangement of a bearing of this type. An oiling drum B is shrunk and keyed on the axle A. This drum revolves in the fixed casing D which the springs and caps C keep pressed against the axlebox E. When the axle revolves in the direction of the arrow X, the drum B forces the oil from the well F into the groove G in the casing D. On leaving through the end of this groove, the oil is forced through the pipes H on to the rubbing surfaces. The superfluous oil is led through the pipe H_1 on to the other side of the bearing, whence it returns to the lubricating device. If the journal revolves in the opposite direction, the oil delivery also changes direction. For the purpose of avoiding leakages of oil the splash rings J_1 and J_2 have been fitted (revolving with the axle) at the points where the axle projects from the box. Oil wastage is prevented by these rings, in conjunction with the box itself, which is of a suitable shape. Protection against dust and water getting in is ensured by a dust shield in several pieces fitted towards the middle of the axle in a pocket in the box; a baffle formed of the splash ring and the box body serve the same purpose. The provision of these bearings on the small locomotives, the wheels of which are only 1.450 m. (4 ft. 9 in.) in diameter, has enabled speeds of 110 to 120 km. (68 to 75 miles) per hour to be reached and maintained.

These bearings lubricated by the forced-feed circulating system necessitated a special construction of the axles, as well as of the wheel centres made of cast steel.

The axles are mounted in the frames in a perfectly symmetrical manner. The easy way in which the engines negociate curves was obtained by the convergence of the two carrying axles, and the choice in this respect was divided between the Adams type and the bissel truck. But as a heavy braking effort was necessary, which required that the carrying axles should also be braked, bissel trucks had to be adopted in which the brake cylinders and brake rigging could be accommodated. The truck centralising arrangement with laminated springs ensures the efficient guiding of the locomotive, both when entering and leaving curves. As a matter of fact, these locomotives are noted for their extremely smooth riding, even at the highest speeds. The main frames and the bissel truck frames are constructed in the usual manner of plates and angles. It may be mentioned here that the bissel truck frames are suspended on the carrying wheels and that the main frame rests on the bissel truck frames without any springs being interposed. The brake hangers for the coupled wheels are equalized by means of angles, so that the frame of the locomotive is carried on three supporting points.

The drawhooks rest on knife edges supported at their sides on frame cross stretchers arranged as near as possible to the coupled pairs of wheels.

The locomotives are fitted with the vacuum brake operating blocks on all the wheels. On the locomotives which were subsequently delivered, the brake is also fitted with a « dead man's » handle arrangement which comes into operation at

the end of a certain distance, and which, in conjunction with the automatic oil firing, enables the engines to be driven by one man only. Like all the other modern locomotives of the Austrian Federal Railways, the locomotive described in the present article is fitted with an exhaust steam pre-heater of the Heinl type, the thermal storage reservoir for which is housed in the lateral water tank on the left side. Besides the two tanks arranged alongside the boiler, a large water tank is also fitted under the coal bunker, right across the locomotive, behind which is placed, also across the frame, the liquid fuel tank.

The driver's cab is situated sufficiently high to give the driver a clear view of the track in either running direction. It adjoins, in the rear, the coal bunker and the luggage compartment and is accessible from the front and rear platforms through four doors. All the fittings and mountings are arranged in the cab in such a manner as to be capable of operation from the fireman's side as well as from the driver's side. This is particularly the case with the reversing gear, the lever for operating the regulator, and the control of the oil firing.

A 500-watt turbo-dynamo ensures the supply of current for lighting, as well as the current required for the head lights fitted at both ends of the locomotive. The coaches which are equipped for electric lighting can also be connected up to this lighting installation.

The small locomotive described above, which has received the designation of « steam rail motor van Class DT.1 » very rapidly proved its worth in service, and was employed on a wide variety of jobs. It was soon used, for example, to haul short fast trains — weighing about 80 t. (78.7 Engl. tons) — on the Semmering line whereon there are 1 in 40 gradients, up which it hauled this load at a speed of 28 km. (17.4 miles) an hour. The indicated horse-power developed on the trial run was as high as 450, with a coal consumption of 1 000 kgr. per m^2 (204 lb. per sq. ft.) of grate area per hour. A particularly noteworthy fact was the high degree of superheat obtained, even for low consumption figures, and this was undoubtedly one of the principal reasons for the extremely economical results given by these locomotives. Under heavier conditions of working, degrees of superheat exceeding 450° C. (842° F.) were obtained.

The great increase in traffic during 1926 and 1927 on the main lines of the Austrian Federal Railways resulted in the 2-D (4-8-0) and 1-C-2 (2-6-4) locomotives of the 113 and 310 classes, exclusively used hitherto, being unable to cope with the express train services. The Federal Railways therefore were compelled to design new and more powerful locomotives. The first point to be settled was the number of coupled pairs of wheels. It had, moreover, to be decided whether the desired reduction in the journey time was to be obtained by greatly increasing the running speed or by rapid acceleration after stopping or slowing down. In the first case, a heavy locomotive with large diameter wheels and three coupled pairs of wheels would be necessary; while in the second a locomotive with wheels of medium diameter and four coupled pairs of wheels would be better. Two locomotives were designed and their timings calculated for the Vienna-Salzburg run.

This showed that the four-coupled engine with rapid acceleration was very much better than the three-coupled, suitable for hauling the trains at high

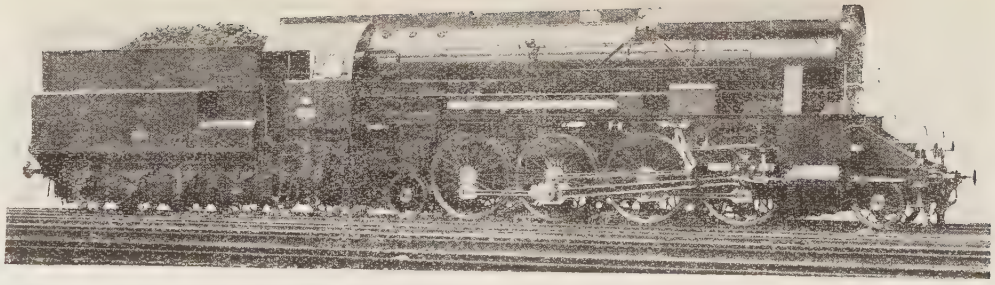
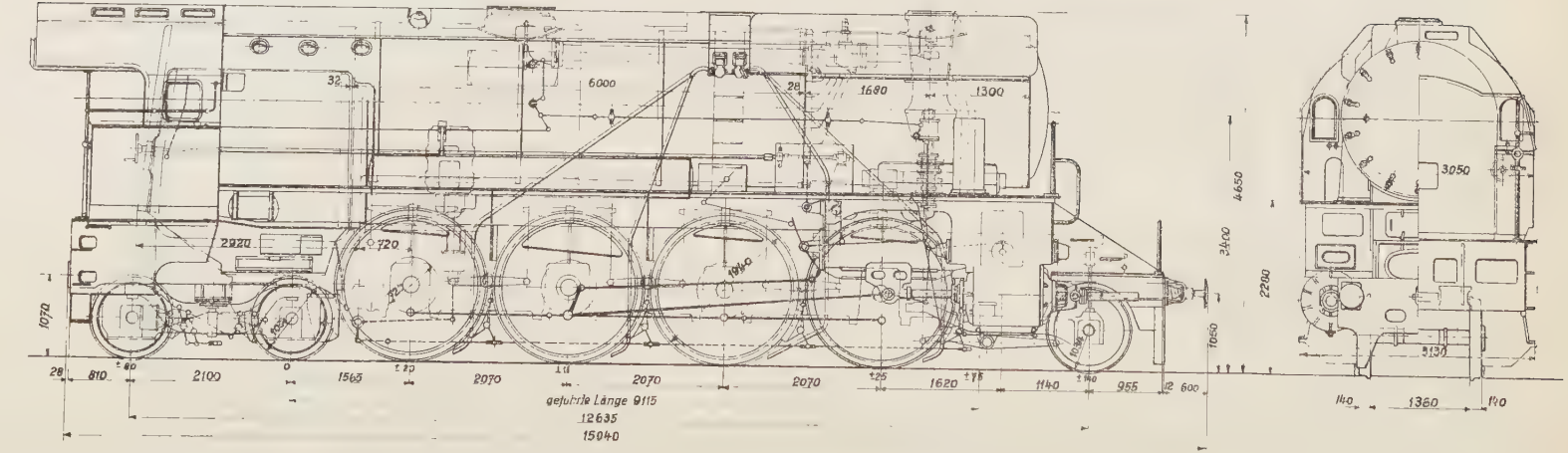
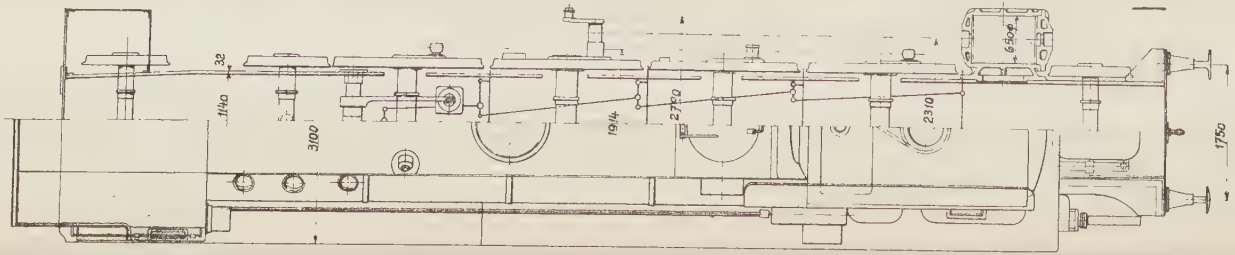


Fig. 4. 2-8-4 express locomotive Class 214, with two simple expansion cylinders, fitted with superheater.





Locomotive data.

Grate area		4.72 m² (50.8 sq. ft.)
Ordinary tubes	Number	147
	Diameter (outside/inside)	57/51.5 mm. (2 1/4" — 2")
Large tubes	Number	38
	Diameter (outside/inside)	143 134 mm. (5 19/32" - 5 9/32")
Heating surface, fireside/waterside	Firebox	18.7/18.8 m² (201.3/202.4 sq. ft.)
	Ordinary tubes	142.6/157.8 m² (1 535/1 699 sq. ft.)
	Large tubes	96.7/102.3 m² (1 062/1 101 sq. ft.)
Evaporating heating surface	Water side	278.9 m² (3 002 sq. ft.)
	Fire side	258.0 m² (2 777 sq. ft.)
Superheating surface	Fire side	91.0 m² (979 sq. ft.)
	Steam side	71.9 m² (774 sq. ft.)
Safety valves, Coale type, 4" Number		2
Effective steam pressure		15 atm. (213.4 lb. per sq. in.)
Diameter of the driving and coupled wheels on tread, tyres 50 mm. (2") thick		1 900 mm. (6' 2 25/32")
Diameter of carrying wheels		994 mm. (3' 3 5/32")
Driving wheels	Diameter of journals	270 mm (10 5/8")
	Length of journals	330 mm. (13")
	Distance centre to centre of journals	1 030 mm. (3' 4 35/64")
Coupled wheels K_1, K_2	Diameter of journals	240 mm. (9 7/16")
	Length of journals	320 mm. (12 9/16")
	Distance centre to centre of journals	1 030 mm. (3' 4 35/64")
Coupled wheels K_3	Diameter of journals	240 mm. (9 7/16")
	Length of journals	310 mm. (12 3/16")
	Distance centre to centre of journals	1 030 mm. (3' 4 35/64")
Carrying wheels	Diameter of journals	200 mm. (7 7/8")
	Length of journals	300 mm. (11 13/16")
	Distance centre to centre of journals	1 060 mm. (3' 5 3/4")
Weight in working order per linear metre (per linear foot), tyres 50 mm. (2"), thick, with length reckoned over the buffers		8.21 t. (8.08 Engl. tons)
Diameter of cylinders		650 mm. 25 5/8")
Piston stroke		720 mm. (28 3/8")
Length of connecting rods		4 250 mm. (13' 11 5/16")

Outside valve gear, type: Walschaerts.
Inside valve gear, type: Lentz.

Empty weight, tyres 50 mm. (2") thick		114.2 t. (112.4 Engl. tons)
Weight fully loaded, tyres 50 mm. (2") thick	1st pair of wheels	16.1 t. (15.8 Engl. tons)
	2nd pair of wheels	18 2 t. (17.9 Engl. tons)
	3rd pair of wheels	18.0 t. (17.7 Engl. tons)
	4th pair of wheels	18.0 t. (17.7 Engl. tons)
	5th pair of wheels	17.94 t. (17.65 Engl. tons)
	6th pair of wheels	17 57 t. (17.29 Engl. tons)
	7th pair of wheels	17.71 t. (17.43 Engl. tons)
Maximum speed		120 km. (75 miles) p. h.
Minimum radius of curve		150 m. (7 1/2 chains)

Automatic compressed air brake, of the Hardy type, with auxiliary brake for the locomotive and tender.

Amplification i' $\frac{1360}{230} \times \frac{546}{366} = 10$ for the coupled wheels.

» $i' = \frac{190}{130}\left\{\frac{360}{160} \times \frac{200}{160} \times \frac{360}{260}\right\} = 6.58$ for the carrying wheels.

Pressure of brake blocks ($\eta = 0.95$)

$\frac{0.95 \times 5.824 \times 10}{8} = 6.91$ t. with $p = 4$ atm. (59.6 lb. per sq. in.)

$\frac{0.95 \times 10.18 \times 10}{8} = 12.08$ t. with $p = 7$ atm. (99.6 lb. per sq. in.)

} for the coupled wheels.

$\frac{0.95 \times 3.120 \times 6.58}{8} = 5.35$ t. with $p = 4$ atm. for the carrying wheels.

Brake percentage of the total weight of the coupled wheels with 4 atm.	76.6 %
Brake percentage of the total weight of the coupled wheels with 7 atm.	134 %
Brake percentage of the total weight of the carrying wheels with 4 atm.	60.7 %
Brake percentage of the total weight in working order with application of the auxiliary brake for the coupled wheels	95.7 %
Pressure of the brake blocks (two 12" cylinders) for the coupled wheels with 4 atm.	5 824 kgr (12 849 lb.)
Pressure of the brake blocks (two 12" cylinders) for the coupled wheels with 7 atm.	10 180 kgr. (22 440 lb.)
Pressure of the brake blocks (one 13" cylinder) for the carrying wheels with 4 atm.	3 420 kgr. (7 540 lb.)

speeds. It should be noted, however, that this superiority only applied to the high-speed line in question, and a special train stopping at several places. These results led to the construction of two trial locomotives, one with three and the other with two simple expansion cylinders, which only differed in the arrangement of the motion. The 1-D-2 (2-8-4) wheel arrangement was adopted in view of the axle load which was limited to 18 tonnes (17.7 Engl. tons), and also the facility with which, with this wheel arrangement, a simple firebox with a flat tube-plate in front, could be employed. As soon as the most important details were settled, the construction of the locomotive was put in hand.

We are only specially concerned in this article with the two-cylinder type (figs. 4 and 5), the type eventually adopted because of its simpler construction and the greater ease with which it could be driven and maintained. Fear was felt at first as to what might happen with peripheral speeds of 13 m. (42.7 ft.) and more per second with the heavy masses of the driving motion necessitated by the pressures of 50 tons on the pistons, but nine years' service have shown that this fear was not justified. As a matter of fact, these locomotives, built to the designs of the Austrian Federal Railways, are distinguished by their extremely smooth running, even at the highest speeds, as well as by the excellent manner in which they can negotiate curves. With regard to the economical efficiency, it may be said that these engines burn the least coal per ton-kilometre of all the Austrian express locomotives. The loads they haul usually exceed 600 t. (590 Engl. tons), and it should be remembered that the Vienna-Salzburg line includes some long climbs (1 in 100-1 in 91). The load which served as a basis when the engine was being designed was that of a train composed of eight-wheeled vehicles, weighing 500 t. (492 Engl. tons) up a 1 in 100 gradient at the speed of 60 km. (37.3 miles) per hour. This programme was easily fulfilled and appreciably exceeded.

We do not propose to enter here into the details of construction of these engines or into the results obtained on the trial runs, because long articles have already been published on the subject in various technical journals, amongst other in this *Bulletin*, and the reproduction of these articles would go beyond the scope of the present article. We shall confine ourselves here to a reference to the improvements embodied in the most recent locomotives, compared with those previously built. In view of the favourable results given in service by the first locomotive, we ordered six of them in 1931 and the same number in 1936.

The method of construction of the boiler has proved satisfactory. In particular it should be noted that the dimensions of the small and large tubes were carefully selected in view of the rate of combustion on the grate, and enable the combustion gases to be utilised to full advantage. Similarly, the dimensions of the smokebox may be considered entirely satisfactory. The superheater, with separate headers for the saturated steam and the superheated steam, has given good results. During intensive express working, the temperature of the steam rises as high as 400° C. (752° F.). The sections of the superheater elements were dimensioned so liberally that with the regulator open and the engine working normally, no appreciable drop in pressure is noted behind the superheater.

The first locomotive of this series is

fitted with a Zara regulator. The locomotive of the second batch had regulators of the Schmidt-Wagner type. The Zara regulator has again been fitted on the most recent locomotives. The reason for this is that large numbers of these Zara regulators are used on the Austrian Federal Railways, where they have proved very satisfactory, and the shops are already familar with their maintenance.

The construction of the cylinders and the valve chests has been modified from time to time. These modifications, however, only apply to details of secondary importance, such as the lubrication, the bye-pass control, the poppet valve spring supports, etc.

The arrangement, outside and inside, of the Lentz poppet valve gear, with oscillating cams, remains unaltered. It would have been useless to make any further improvement in them, since the conditions afforded for the admission and exhaust by the poppet valve gear enable the locomotive to work with cut-offs of 12 % without any unstability effect on its running.

The frame, axleboxes, and the pairs of wheels were completely remodelled, in view of the decision taken to fit the locomotive with bearings with circulating lubrication. The first engines have bearings oiled from mechanical (forced feed) lubricators with hose delivery pipes. Owing to trouble with these hose pipes on several occasions, it was decided to adopt the circulating oil system of the Friedmann type, which has been employed with very good results on steam rail motor vans. As this type of lubrication involved bearings of larger dimensions, the frames, bissel truck and wheels had to be completely re-constructed. Needless to say, the same principles of construction were followed as for the locomotives previously delivered. The bearings are of the same shape as those described above fitted on the steam rail motor vans. We may mention in this respect that the method of construction of the main frame of these engines, with frame plates and box type cross stretchers in steel castings, has given very good results. Several locomotives have met with serious accidents and overturned after being derailed; the frame stood up to this very well and only had to undergo light repairs.

For the purpose of accommodating the lubricating rings on the wheel centres, it was necessary to dish in these latter to a certain extent, and consequently, the axles had to be made longer than on the locomotives with forced-feed lubrication.

The connecting and coupling rods, as well as all the valve motion rods proved to be so very satisfactory in service on the first locomotives, that scarcely any modifications were made to them. At the beginning, all the brasses for the rods of these locomotives were lined with white metal, of only 5 mm. (13/64 in.) thickness. Up to the present, after about five months service, during which the locomotives have covered on the average 10 000 km. (6 250 miles) per month, this arrangement has given very good results.

The rear bogie with a fixed pivot above the leading pair of wheels has also been retained without any change. When the first three-cylinder simple expansion locomotive, mentioned above, was built, doubts were felt with regard to its running qualities, and the first two-cylinder locomotive was fitted with a central bogie having side play. But in service the locomotive with the fixed pivot above the leading pair of wheels proved to have better running qualities than that with

the central pivot, and consequently all the locomotives ordered from the second batch were equipped with bogies like that fitted on the three-cylinder locomotive.

The engines are equipped with feed-water pre-heaters of the Heinl type, with thermal storage reservoirs in the tender water tank. This pre-heater is no different in principle from the equipment of the same type employed on the steam rail motor vans.

From the point of view of appearance, the principal difference between the recently delivered locomotives and those built previously is the shape of the wind deflector plates. The deflectors of the old type, fixed on both sides of the footboard, had the disadvantage of interfering with the driver's view from the cab. A trial was first made, on the new locomotives, with a deflector consisting of a side cowling carried forward from the dome to the front of the chimney, but the results were far from satisfactory. After many trials, it was decided to adopt the shape of deflector shewn in the figure, and this has proved satisfactory in service.

The locomotives are fitted with the compressed air brake. The vacuum brake, which was still in use on the old locomotives of the same type for braking the train, was abandoned in favour of the compressed air brake, which has come more and more into general use on the Austrian Federal Railways.

Like all modern locomotives, the engines described above are fitted with all modern mountings and fittings, such as electric light, etc., so as to reduce to a minimum the labour demanded of the enginemen. Particular care was taken, in the construction, to make all the parts which demand periodical examination easy of access and operation. It is therefore not surprising that these locomotives are very much liked by the enginemen, which is, generally, an essential condition in the efficient operation of locomotives.

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The stability of welded track,

by W. J. VAN DER EB,

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The adoption of very long lengths of welded rails has given rise to the problem of the stability of the track. The Dutch East Indies State Railways, having the use of such rails in view, instructed the author of the present paper to investigate the subject, and the results show that the stability of a section of permanent way can be rapidly verified by means of a table, or a graph.

These results are of a general character and allow of the buckling of the track being investigated on the assumption of the presence of any particular number of waves. The solution depends upon two factors, represented hereafter by the letters γ and δ, appropriate to a known example of buckling. The present paper gives these factors for four cases of buckling, viz : — on the assumption of the formation of 1, 2, 3, or an infinite number of waves. These factors can be determined for any number of waves, but the author has confined his researches to the cases quoted, practice having shown that an instance of buckling with more than three waves has not yet been observed.

It is proposed to examine in turn :

(1) The influences tending to promote stability;

(2) The effect of longitudinal frictional resistance;

(3) The moment of inertia to be adopted;

(4) The buckling action;

(5) The stability of the permanent way on curves.

CHAPTER I.

Influences favourable to stability.

Buckling in the vertical plane is opposed by the weight of the track.

To the reaction of the dead weight which is uniformly distributed, must be added the frictional resistance offered by any tendency to vertical movement of the sleepers in the ballast.

As they rise, the amount of their surface in contact with the ballast decreases with a corresponding decrease in the frictional resistance,

The frictional resistance is therefore not constant and being relatively small, we shall neglect it, thus simplifying the problem.

Horizontal buckling is opposed by the sleepers being surrounded by ballast and the Permanent Way Engineer's Department of the Dutch East Indies State Railways has conducted experiments to investigate this aspect of the matter, with the object of arriving at the amount of resistance offered by the ballast to the sleepers slipping sideways, varying with the vertical displacement stress, both for stone and sand ballast.

Figures 1, 2, and 3 show the results obtained, and it will be noted that *the resistance arising from the surrounding ballast, at least when load is first ap-*

Stone ballast. — Wooden sleepers. — Size of sleepers : 200 × 22 × 12 cm. $\left(6'6\ \frac{3''}{4} \times 8\ \frac{11''}{16} \times 1\ \frac{3''}{4}\right)$.

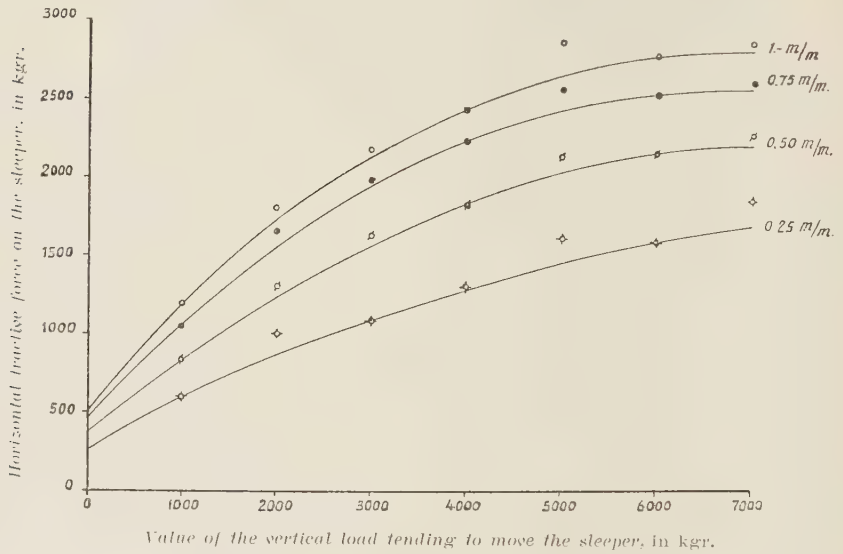


Fig. 1.

Sand ballast. — Wooden sleepers. — Size of sleepers : 200 × 22 × 12 cm.

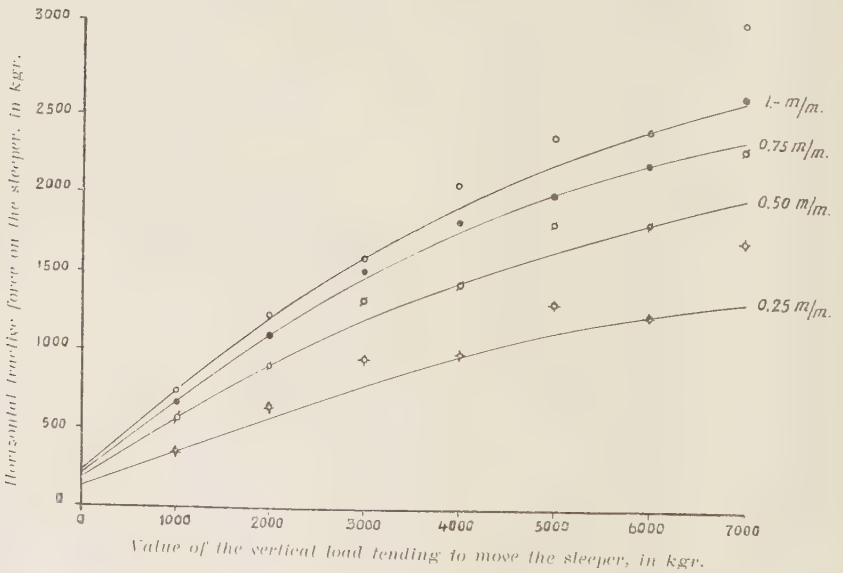


Fig. 2.

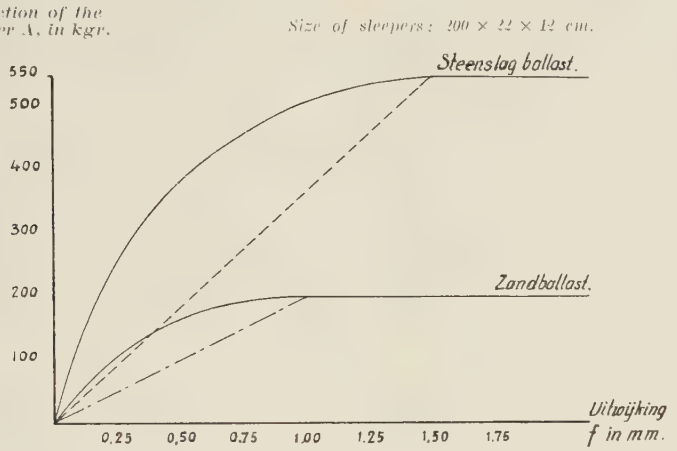


Fig. 3. — Characteristics of the sleeper and ideal characteristics without vertical load tending to move the wooden sleepers.

Explanation of Dutch terms:
Steenslagballast = broken stone ballast. — Zandballast = sand ballast. — Uitwijking *f*, in mm. = displacement *f*, in mm.

plied to the track, is not constant but is in some measure proportional to the displacement. Only after a certain value of the latter does the resistance tend to become constant.

CHAPTER II.

The effect of longitudinal frictional resistance

The resistance to longitudinal slipping depends on two factors, viz :

(1) Slipping movement in the ballast;

(2) The slipping of the track in its fastenings.

The effect of the first has been dealt with in Chapter 1.

The frictional resistance offered by the sole plates has been determined experimentally for the three types of track now in use, the average figures being :

Track laid with No. 2 type rails, 0.49 tonne (0.482 Engl. ton).

Track laid with No. 3 type rails, 0.85 tonne (0.846 Engl. ton).

Track laid with No. 14 type rails, 2.24 tonnes (2.205 Engl. tons).

It is advisable to add a certain co-efficient of safety to these figures in case some fastenings are not sufficiently tight. In the example given of calculations concerning No. 3 type rails and fastenings, the figure of 0.25 tonne (0.246 Engl. ton) has been assumed for longitudinal resistance.

CHAPTER III.

Moment of inertia to be adopted

For buckling in a vertical plane, twice the moment of inertia of a rail around its horizontal axis should be taken and the effect of the sole plates neglected.

The question of the horizontal buckling is a more delicate one, the track forming a sort of *Vierendeel* girder, the joints of which being more or less rigid. The value of the moment to be admitted can only be arrived at by experiments, such as those carried out by the State Railways. These experiments consisted in measuring the deflection taken by a section of track submitted to the effect of its dead weight, with the two following assumptions :

(1) All rail fastenings perfectly tigh

(2) Fastenings eased to some extent.

The following table of the results ob tained shows the great influence of th tightness of the fastenings. When th latter are eased, deflection takes place if there were no sleepers, the moment inertia of the whole being equal to twi that of a rail around its vertical axis inertia. The effect of the fastenings only felt if the deflection increases.

Rail section. Type.	F in cm².	I_x in cm⁴.	I_y in cm⁴.	Deflection measured in cm. with		To be included in the term I_h in cm⁴, in the case of		No. of sleepers.	Sleeper spacing, cm.
				tight fastenings.	eased fastenings.	tight fastenings.	eased fastenings.		
Rail No. 2	33	538	89	13.4	35.5	440	180	13	31
Rail No. 3	42.5	1 037	151	19.8	86.5	1 500	340	20	69
Rail No. 14	53.1	1 360	235	21.2	61.5	1 500	520	20	69

CHAPTER IV.

Deformation and buckling of the track.

As soon as buckling appears the resistance to slipping offered by the sleepers gives rise to reactions of a constant value, which tend to oppose any slip.

The track buckles in several waves, each section of which needs examining separately.

The following phenomenon is observed : in a given section the equilibrium of the track is upset and the track thrown sideways; the chord of that section shortens while the adjacent sections become lengthened over a certain distance, determined by the frictional resistance of the fastenings.

Supposing the track of identical c struction throughout, the conditions p duced in the adjacent lengths of line be identical. Beyond the three leng mentioned the stress due to the a compression of the rails persists.

The question is to determine the m nitude of this restraining force wh we call P_t, P being the axial force in buckled portion, and always less than

The variation from P to P_t being sumed to be linear, with the objec treating the question in a general fash let us consider any particular wave lected which we find subjected to following forces :

An axial force P, a uniformly di buted transverse load q, two any ments M_1 and M_2, and two any tr

verse forces Q_1 and Q_2, at the ends of the wave (see fig. 4).

The equation of the curve of elasticity is given by

$$E \cdot I \cdot \frac{d^2y}{dx^2} = + M_x$$

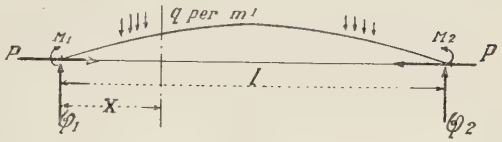


Fig. 4.

while

$$M_x = - P \cdot y + \tfrac{1}{2} \cdot q \cdot x \cdot (l - x) + \frac{l - x}{l} \cdot M_1 + \frac{x}{l} \cdot M_2.$$

The differential equation thus becomes :

$$\frac{d^2y}{dx^2} = - a^2 \cdot y + \frac{\tfrac{1}{2} \cdot q \cdot x \cdot (l - x) + \frac{l - x}{l} \cdot M_1 + \frac{x}{l} \cdot M_2}{E \cdot I}$$

if :

$$a^2 = \frac{P}{E \cdot I}.$$

The integral of this equation is :

$$y = C_1 \cdot \cos ax + C_2 \cdot \sin ax + \frac{\tfrac{1}{2} \cdot q \cdot x \cdot (l - x) + \frac{l - x}{l} \cdot M_1 + \frac{x}{l} \cdot M_2}{P} + \frac{q}{a_2 \cdot P}.$$

and consequently :

$$\frac{dy}{dx} = - C_1 \cdot a \cdot \sin ax + C_2 \cdot a \cdot \cos ax + \frac{\tfrac{1}{2} \cdot q \cdot l - q \cdot x - \frac{M_1}{l} + \frac{M_2}{l}}{P}.$$

Given $y = 0$ for $x = 0$ and $x = l$, we get the two equations for C_1 and C_2 :

$$C_1 + \frac{M_1}{P} + \frac{q}{a^2 \cdot P} = 0$$

and

$$C_1 \cdot \cos al + C_2 \cdot \sin al + \frac{M_2}{P} + \frac{q}{a^2 \cdot P} = 0$$

from which we get :

$$C_1 = - \frac{a^2 \cdot M_1 + q}{a^2 \cdot P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$C_2 = + \frac{a^2 \cdot M_1 + q}{a^2 \cdot P} \cdot \operatorname{cotan} al - \frac{a^2 \cdot M_2 + q}{a^2 \cdot P \cdot \sin al} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Let us now consider the first case, i.e. a deformation following a single wave : In this case $Q_1 = Q_2 = 1/2\ q.l$, while $M_1 = M_2$ and $\frac{dy}{dx} = 0$ for $x = 0$ (see fig. 5).

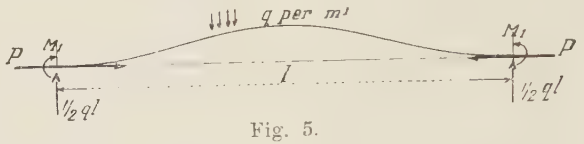


Fig. 5.

The last condition gives a third equation for $M_1 = M_2$, i.e. :

$$C_2 \cdot a + \frac{q \cdot l}{2 \cdot P} = 0$$

or

$$C_2 = -\frac{q \cdot l^2}{2 \cdot P \cdot al}.$$

Using (1) and (2) we get : $C_1 = \frac{q \cdot l^2}{2 \cdot P} \cdot \frac{\sin al}{al \cdot (1 - \cos al)}$

and

$$M_1 = \frac{q \cdot l_2}{2 \cdot al} \cdot \frac{\sin al}{1 - \cos al} - \frac{q}{a^2}$$

whence :

$$y = -\frac{q \cdot l^2}{2 \cdot P} \cdot \left[\frac{\sin al \cdot (\cos ax - l)}{al \cdot (1 - \cos al)} + \frac{\sin ax}{al}\right] + \frac{\frac{1}{2} \cdot q \cdot x \cdot (l - x)}{P} \quad . \quad (3)$$

The deflection becomes :

$$f = \frac{q \cdot l}{2 \cdot a \cdot P} \cdot \left[\frac{al}{4} - \tan \frac{al}{4}\right]$$

$$\frac{q \cdot E \cdot I}{2 \cdot P^2 \cdot f} = \frac{1}{al \cdot \left(\frac{al}{4} - \tan \frac{al}{4}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

Endeavouring to find at what value of l, q becomes maximum, P and f remaining constant, we get, differentiating with regard to l :

$$\frac{dq}{dl} = \frac{16 \cdot a \cdot P \cdot f}{l^2} \cdot \frac{(\cos \frac{1}{2} al - 1)\,(al \cdot \cos \frac{1}{2} al \quad 2 \cdot \sin \frac{1}{2} al)}{al \cdot \sin \frac{1}{2} al - 4 \cdot (1 - \cos \frac{1}{2} al)}$$

an expression which has to be cancelled out by making the denominator infinite or the numerator zero.

The denominator becomes $\sim$ if $al = \sim$ which has no practical significance. The numerator becomes 0 if $\cos \frac{1}{2} al = 1$ or $\frac{al}{2} = \tan \frac{al}{2}$.

The smallest value of al, resulting from the second condition, is : $al = 2.86\,\pi = 9$ approximately,

whence :
$$P = \frac{81 \cdot E \cdot I}{l^2} \quad \ldots \ldots \ldots \ldots \quad (4)$$

and :
$$q = 0.064 \cdot \frac{P^2 \cdot f}{E \cdot I} \quad \ldots \ldots \ldots \ldots \quad (5)$$

Let us now determine the shortening of the chord, due to the deflection; we get :
$$\Delta_1 l = \tfrac{1}{2} \int_0^l \left(\frac{dy}{dx}\right)^2 dx.$$

$$\Delta_1 l = \int_0^l \left[\tfrac{1}{2} \cdot C_1^2 \cdot a^2 \cdot \sin^2 ax + \tfrac{1}{2} \cdot C_2^2 \cdot a^2 \cos^2 ax + \frac{\tfrac{1}{4} \cdot q^2 \cdot l^2 + q^2 \cdot x^2 - q^2 \cdot l \cdot x}{2 \cdot P^2} - \right.$$

$$- C_1 \cdot C^2 \cdot a^2 \cdot \sin ax \cdot \cos ax - C_1 \cdot a \cdot \sin ax \, \frac{\tfrac{1}{2} \cdot q \cdot l - q \cdot x}{P} + C_2 \cdot a \cdot \cos$$

$$\left. ax \cdot \frac{\tfrac{1}{2} \cdot q \cdot l - q \cdot x}{P}\right] dx = \tfrac{1}{2} \cdot C_1^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot al - \tfrac{1}{4} \cdot \sin 2\,al\right) + \tfrac{1}{2} \cdot C_2^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot al + \right.$$

$$\left. + \tfrac{1}{4} \cdot \sin 2\,al\right) - \tfrac{1}{2} \cdot C_1 \cdot C_2 \cdot a \cdot \sin^2 al + \frac{C_1 \cdot q \cdot l}{2 \cdot P} \cdot (\cos al - 1) + \frac{C_2 \cdot q \cdot l}{2 \cdot P}$$

$$\sin al + \frac{C_1 \cdot q}{a \cdot P} \cdot (\sin al - al \cdot \cos al) - \frac{C_2 \cdot q}{a \cdot P} (\cos al - 1 + al \cdot \sin al) +$$

$$+ \frac{q^2 \cdot l^3}{24 \cdot P^2} = 76 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\dfrac{P}{E \cdot I}}} \quad \ldots \ldots \ldots \ldots \quad (6)$$

substituting 9 for al.

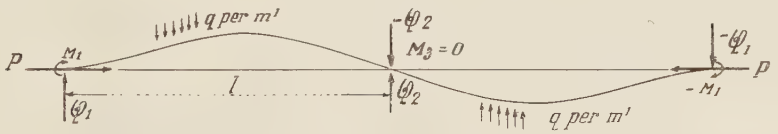


Fig. 6.

Second example (see fig. 6).

In this case there are two waves to be considered. Limiting ourselves to the study of one of them, the other being similar to it, we have $\frac{dy}{dx} 0$ for $x = 0$, while $M_2 = 0$.

Whence :
$$C_2 \cdot a + \frac{\tfrac{1}{2} \cdot q \cdot l^2 - M_1}{P \cdot l} = 0.$$

Using (1) and (2) we find :

$$C_1 = -\frac{a^2 \cdot M_1 + q}{a^2 \cdot P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$C_2 = \frac{a^2 \cdot M_1 + q}{a^2 \cdot P} \cdot \operatorname{cotan} al - \frac{q}{a^2 \cdot P \cdot \sin al} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

änd $$M_1 = -q \cdot l^2 \cdot \left[\frac{2 \cdot (\cos al - 1) + al \cdot \sin al}{2 \cdot al \cdot (al \cdot \cos al - \sin al)}\right] = -\Delta \cdot q \cdot l^2 \quad . \quad (9)$$

Substituting $x = \frac{l}{2}\, l$, we get :

$$f = \frac{q}{a^2 \cdot P} \cdot \left[1 - \frac{1}{\cos \frac{al}{2}} + \frac{a^2 l^2}{8}\right] + \frac{M_1}{2 \cdot P} \cdot \left(1 - \frac{1}{\cos \frac{al}{2}}\right) \quad . \quad (10)$$

In solving the equation (10), q has to attain a relative maximum, and we find $al = 6.4$ is the most unfavourable value :

$$P = \frac{41 \cdot E \cdot I}{l^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and $$q = 0.148\, \frac{P^2 \cdot f}{E \cdot I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

whence

$$\Delta_1 l = \int_0^l \left[\tfrac{1}{2} \cdot C_1^2 \cdot a^2 \cdot \sin^2 ax + \tfrac{1}{2} \cdot C_2^2 \cdot a^2 \cdot \cos^2 ax + \tfrac{1}{2} \cdot \left(\frac{\tfrac{1}{2} \cdot q \cdot l - q \cdot x - \frac{M_1}{l}}{P}\right)^2 - \right.$$

$$- C_1 \cdot C_2 \cdot a^2 \cdot \sin ax \cdot \cos ax - C_1 \cdot a \cdot \sin ax \cdot \frac{\tfrac{1}{2} \cdot q \cdot l - q \cdot x - \frac{M_1}{l}}{P} + C_2 \cdot a \cdot \cos ax \cdot$$

$$\left. \cdot \frac{\tfrac{1}{2} \cdot q \cdot l - q \cdot x - \frac{M_1}{l}}{P}\right] dx = \tfrac{1}{2} \cdot C_1^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot al - \tfrac{1}{4} \sin 2\, al\right) + \tfrac{1}{2} \cdot C_2^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot al + \right.$$

$$+ \tfrac{1}{4} \cdot \sin 2\, al) - \frac{C_1 \cdot C_2 \cdot a}{2} \cdot \sin^2 al + \frac{C_1 \cdot \left(q \cdot l - \frac{2 \cdot M_1}{l}\right)}{2 \cdot P} (\cos al - 1) +$$

$$\left. + \frac{C_2 \cdot \left(q \cdot l - \frac{2 \cdot M_1}{l}\right)}{2 \cdot P}\right) \cdot \sin al + \frac{C_1 \cdot q}{a \cdot P} \cdot (\sin al - al \cdot \cos al) - \frac{C_2 \cdot q}{a \cdot P} \cdot$$

$$\cdot (\cos al - 1 + al \cdot \sin al) + \frac{q^2 \cdot l^3}{24 \cdot P^2} + \frac{M_1^2}{2 \cdot P^2 \cdot l} = 34 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}} \quad . \quad . \quad (13)$$

Therefore, if q reaches a maximum, the axial force becomes equal to twice the buckling load, according to EULER.

Third example (see fig. 7).

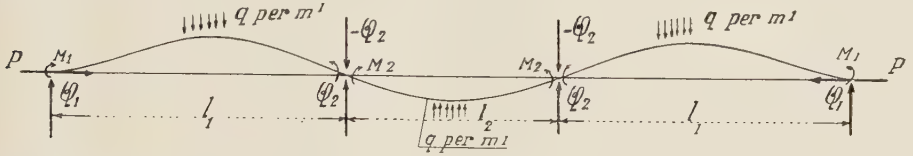


Fig. 7.

For the first wave $\frac{dy}{dx} = 0$ for $x = 0$, therefore :

$$0 = C_2\, a + \frac{\frac{1}{2} q \cdot l_1 - \frac{M_1}{l_1} + \frac{M_2}{l_1}}{P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Using (1) and (2) we find that :

$$C_1 = \frac{q \cdot l_1^2}{P} \cdot \left(\Delta_1 - \frac{1}{a^2\, l_1^2}\right) - \frac{M_2}{P} \cdot \Delta_2$$

$$C_2 = -\frac{q \cdot l_1^2}{P} \cdot \left\{\frac{2 \cdot \Delta_1 + 1}{2a\, l_1}\right\} + \frac{M_2}{P} \cdot \left\{\frac{\Delta_2 - 1}{a\, l_1}\right\}$$

$$M_1 = -\Delta_1 \cdot q \cdot l_1^2 + \Delta_2 \cdot M_2$$

assuming :

$$\Delta_1 = \frac{2 \cdot (\cos a\, l_1 - 1) + a\, l_1 \cdot \sin a\, l_1}{2 \cdot a\, l_1 \cdot (a\, l_1 \cdot \cos a\, l_1 - \sin a\, l_1)}$$

$$\Delta_2 = \frac{\sin a\, l_1 - a\, l_1}{\sin a\, l_1 - a\, l_1 \cdot \cos a\, l_1}$$

whence we have :

$$\left(\frac{dy}{dx}\right)_{x=l_1} = -\frac{q \cdot l_1}{P} \cdot \left[\left(\Delta_1 - \frac{1}{a^2\, l_1^2}\right) \cdot a\, l_1 \cdot \sin a\, l_1 + \left(\frac{2 \cdot \Delta_1 + 1}{2}\right) \cdot \cos a\, l_1 + \right.$$

$$\left. + \left(\tfrac{1}{2} - \Delta_1\right)\right] + \frac{M_2}{P \cdot l_1} \cdot \left[\Delta_2 \cdot a\, l_1 \cdot \sin a\, l_1 + (\Delta_2 - 1) \cos a\, l_1 + (1 - \Delta_2)\right] =$$

$$-\frac{q \cdot l_1}{P} \cdot Y_1 + \frac{M_2}{P \cdot l_1} \cdot Y_2.$$

For the second wave $M_1 = M_2$, therefore :

$$C_1 = -\frac{M_2 + \frac{q}{a^2}}{P}$$

$$C_2 = \frac{q}{a^2 \cdot P} \cdot \left(\frac{\cos al_2 - 1}{\sin al_2}\right) + \frac{M_2}{P} \cdot \left(\frac{\cos al_2 - 1}{\sin al_2}\right) \quad . \quad . \quad . \quad . \quad (15)$$

and :

$$\left(\frac{dy}{dx}\right)_{x=0} = C_2 a + \frac{q \cdot l_2}{2 \cdot P} = \frac{q \cdot l_2}{P} \cdot \left(\frac{\cos al_2 - 1}{al_2 \cdot \sin al_2} + \frac{1}{2}\right) + \frac{M_2}{P \cdot l_2} \cdot$$

$$\cdot \frac{al_2 \cdot (\cos al_2 - 1)}{\sin al_2} = \frac{q \cdot l_2}{P} \cdot Y_3 + \frac{M_2}{P \cdot l_2} \cdot Y_4.$$

Giving M_2 a positive value for the first wave, it is necessary to give the second a corresponding negative value, whereas :

$$\left(\frac{dy}{dx}\right)_{x=l_1} = -\left(\frac{dy}{dx}\right)_{x=0}.$$

Whence $$-\frac{q \cdot l_1}{P} \cdot Y_1 + \frac{M_2}{P \cdot l_1} \cdot Y_2 = -\frac{q \cdot l_2}{P} \cdot Y_3 + \frac{M_2}{P \cdot l_2} \cdot Y_4$$

or $$M_2 = q \cdot \frac{l_1 \cdot l_2 \cdot (Y_1 \cdot l_1 - Y_3 \cdot l_2)}{(Y_2 \cdot l_2 - Y_4 \cdot l_1)}.$$

Assuming that $M_2 = \alpha \cdot q \cdot {l_2}^2$, we find :

$$\alpha = \frac{l_1}{l_2} \cdot \frac{Y_1 \cdot l_1 - Y_3 \cdot l_2}{Y_2 \cdot l_2 - Y_4 \cdot l_1} = \frac{al_1}{al_2} \cdot \frac{al_1 \cdot Y_1 - al_2 \cdot Y_3}{al_2 \cdot Y_2 - al_1 \cdot Y_4} \quad . \quad . \quad . \quad (16)$$

For the second wave, we get for $x = \frac{1}{2} l_2$:

$$f = \frac{q}{a^2 \cdot P} \cdot \left[1 - \frac{1}{\cos \frac{al_2}{2}} + \frac{a^2 {l_2}^2}{8}\right] + \frac{M_2}{P} \cdot \left[1 - \frac{1}{\cos \frac{al_2}{2}}\right] \quad . \quad . \quad (17)$$

Solving equation (16), the value of q having to attain a relative maximum if P and f remain constant, we find $al_1 = \sim 5.8$ and $al_2 = \sim 4.2$, as the most unfavourable figures.

Continuing, we have :

$$P = \frac{33 \cdot 6 \cdot E \cdot}{{l_1}^2} = \frac{17.6 \cdot E \cdot I}{{l_2}^2} \quad \text{whereas} \quad q = 0.111 \cdot \frac{P^2 \cdot f^2}{E \cdot I}.$$

For the first wave :

$$\Delta_1 l_1 = \tfrac{1}{2} \cdot \int_0^{l_1} \left[-C_1 \cdot a \cdot \sin ax + C_2 \cdot a \cdot \cos ax + \frac{\tfrac{1}{2} \cdot q \cdot l_1 - q \cdot x - \dfrac{M_1}{l_1} + \dfrac{M_2}{l_1}}{P} \right]^2 dx =$$

$$= \tfrac{1}{2} \cdot C_1^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot a l_1 - \tfrac{1}{4} \cdot \sin 2\, a l_1\right) + \tfrac{1}{2} \cdot C_2^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot a l_1 + \tfrac{1}{4} \cdot \sin 2\, a l_1\right) -$$

$$- \tfrac{1}{2} \cdot C_1 \cdot C_2 \cdot a \cdot \sin^2 a l_1 + \frac{C_1 \cdot \left[q \cdot l_1 + 2 \cdot \left(\dfrac{M_2 - M_1}{l_1}\right)\right]}{2 \cdot P} \cdot (\cos a l_1 - 1) +$$

$$+ \frac{C_2 \cdot \left[q \cdot l_1 + 2 \cdot \left(\dfrac{M_2 - M_1}{l_1}\right)\right]}{2 \cdot P} \cdot \sin a l_1 + \frac{C_1 \cdot q}{a \cdot P} \cdot (\sin a l_1 - a l_1 \cos a l_1) - \frac{C_2 \cdot q}{a \cdot P} \cdot$$

$$\cdot (\cos a l_1 - 1 + a l_1 \cdot \sin a l_1) + \frac{q^2 \cdot l_1^3}{24 \cdot P^2} + \frac{(M_2 - M_1)^2}{2 \cdot P^2 \cdot l_1} = 23 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\dfrac{P}{E \cdot I}}}$$

and for the second :

$$\Delta_1 l_2 = \tfrac{1}{2} \int_0^{l_2} \left[-C_1 \cdot a \cdot \sin ax + C_2 \cdot a \cdot \cos ax + \frac{\tfrac{1}{2} \cdot q \cdot l_2 - q \cdot x}{P} \right]^2 dx =$$

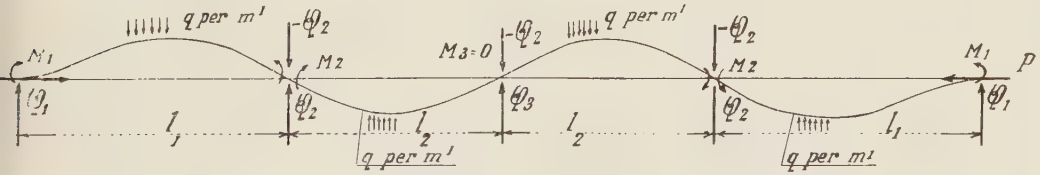
$$= \tfrac{1}{2} \cdot C_1^2 \cdot a \cdot \left(\tfrac{1}{2} \cdot a l_2 - \tfrac{1}{4} \cdot \sin 2\, a l_2\right) + \tfrac{1}{2} \cdot C_2^2 \cdot a \cdot \left(\tfrac{1}{2} a l_2 + \tfrac{1}{4} \cdot \sin 2\, a l_2\right) -$$

$$- \tfrac{1}{2} \cdot C_1 \cdot C_2 \cdot a \cdot \sin^2 a l_2 + \frac{C_1 \cdot q \cdot l_2}{2 \cdot P} \cdot (\cos a l_2 - 1) + \frac{C_2 \cdot q \cdot l_2}{2 \cdot P} \cdot \sin a l_2 +$$

$$+ \frac{C_1 \cdot q}{a \cdot P} \cdot (\sin a l_2 - a l_2 \cdot \cos a l_2) - \frac{C_2 \cdot q}{a \cdot P} \cdot (\cos a l_2 - 1 + a l_2 \cdot \sin a l_2) +$$

$$+ \frac{q^2 \cdot l_2^3}{24 \cdot P^2} = 46 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\dfrac{P}{E \cdot I}}}.$$

Fourth example (see fig. 8).



Note. — In this figure, instead of : — Q_2 (opposed to Q_3), read : — Q_3.

Fig. 8.

For the first wave we will refer to the third example and consider the second wave directly. The moment of the latter to the right is zero.

Representing the moment to the left M_1 by M_2, we get :

$$\left(\frac{dy}{dx}\right)_{x=0} = \frac{q \cdot l^2}{P} \cdot \left[\frac{\cos al_2 - 1}{al_2 \cdot \sin al_2} + \frac{1}{2}\right] + \frac{M_2}{P \cdot l_2} \cdot [al_2 \cdot \operatorname{cotan} al_2] =$$

$$= \frac{q \cdot l_2}{P} \cdot Y_3^1 + \frac{M_2}{P \cdot l_2} \cdot Y_4^1.$$

Assuming that $M = \alpha \cdot q \cdot l_2^2$, we have similarly :

$$\alpha = \frac{al_1}{al_2} \cdot \frac{al_1 \cdot Y_1 - al_2 \cdot Y_3^1}{al_2 \cdot Y_2 - al_1 \cdot Y_4^1}.$$

And then :

$$f = \frac{q \cdot E \cdot I}{P^2} \cdot \left[\left\{1 - \frac{1}{\cos \frac{al_2}{2}} + \frac{a^2 \, l_2^2}{8}\right\} + \frac{\alpha}{2} \cdot a^2 \, l_2^2 \left\{1 - \frac{1}{\cos \frac{al_2}{2}}\right\}\right]. \quad (18)$$

In this case we have : $a \, l_1 = \sim 5.3$ and $al_2 = \sim 4.2$.

Furthermore :

$$P = \frac{28.1 \cdot E \cdot I}{l_1^2} = \frac{17.6 \cdot E \cdot}{l_2^2} \text{ while } q = 0.116 \cdot \frac{P^2 \cdot f_2}{E \cdot I}.$$

In addition, we have, for the first wave :

$$\Delta_1 l_1 = \frac{1}{2} \int_0^{l_1} \left(-C_1 \cdot a \cdot \sin ax + C_2 \cdot a \cdot \cos ax + \frac{\frac{1}{2} \cdot q \cdot l_1 - q \cdot x - \frac{M_1}{l_1} + \frac{M_2}{l_1}}{P}\right)^2 dx =$$

$$= \frac{1}{2} \cdot C_1^2 \cdot a \cdot \left(\frac{1}{2} \cdot a \, l_1 - \frac{1}{4} \cdot \sin 2a \, l_1\right) + \frac{1}{2} \cdot C_2^2 \cdot a \cdot \left(\frac{1}{2} \cdot a \, l_1 + \frac{1}{4} \cdot \sin 2a \, l_1\right) -$$

$$- \frac{1}{2} \cdot C_1 \cdot C_2 \cdot a \cdot \sin^2 a \, l_1 + \frac{C_1 \cdot \left[q \cdot l_1 + 2 \cdot \left(\frac{M_2 - M_1}{l_1}\right)\right]}{2 \cdot P} \cdot (\cos a \, l_1 - 1) +$$

$$+ \frac{C_2 \cdot \left[q \cdot l_1 + 2 \cdot \left(\frac{M_2 - M_1}{l_1}\right)\right]}{2 \cdot P} \cdot \sin a \, l_1 + \frac{C_1 \cdot q}{a \cdot P} \cdot (\sin a \, l_1 - a \, l_1 \cos a \, l_1) -$$

$$- \frac{C_2 \cdot q}{a \cdot P} \cdot (\cos a \, l_1 - 1 + a \, l_1 \cdot \sin a \, l_1) + \frac{q^2 \cdot l_1^3}{24 \cdot P^2} + \frac{(M_2 - M_1)^2}{2 \cdot P^2 \cdot l_1} = 11 \, \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}} =$$

and for the second wave

$$\Delta_1 l_2 = \int_0^{l_2} \left[\tfrac{1}{2} \cdot C_1^2 \cdot a^2 \cdot \sin^2 ax + \tfrac{1}{2} \cdot C_2^2 \cdot a^2 \cdot \cos^2 ax + \tfrac{1}{2} \cdot \left(\frac{\tfrac{1}{2} \cdot q \cdot l_2 - q \cdot x - \dfrac{M_2}{l_2}}{P} \right)^2 - \right.$$

$$- C_1 \cdot C_2 \cdot a^2 \cdot \sin ax \cdot \cos ax - C_1 \cdot a \cdot \sin ax \cdot \frac{\tfrac{1}{2} \cdot q \cdot l_2 - q \cdot x - \dfrac{M_2}{l_2}}{P} + C_2 \cdot a \cdot \cos ax \cdot$$

$$\left. \cdot \frac{\tfrac{1}{2} \quad q \cdot l_2 - q \quad x - \dfrac{M_2}{l_2}}{P} \right] dx = \tfrac{1}{2} \cdot C_1^2 \cdot a \cdot (\tfrac{1}{2} \cdot a l_2 - \tfrac{1}{4} \cdot \sin^2 a l_2) + \tfrac{1}{2} \cdot C_2^2 \cdot a \cdot (\tfrac{1}{2} \cdot a l_2 +$$

$$+ \tfrac{1}{4} \quad \sin^2 \; a \; l_2) - \frac{C_1 \cdot C_2 \, a}{2} \cdot \quad \sin^2 a \; l_2 + \frac{C_1 \cdot \left(q \cdot l_2 - \dfrac{2 \cdot M_2}{l_2} \right)}{2 \cdot P} (\cos \; a \; l_2 \; - \; 1) +$$

$$+ \frac{C_2 \cdot \left(q \cdot l_2 - \dfrac{2 \cdot M_2}{l_2} \right)}{2 \cdot P} \cdot \quad \sin \, a \, l_2 + \frac{C_1 \cdot q}{a \cdot P} \cdot (\sin \, a \, l_2 - a \, l_2 \cdot \cos \, a \, l_2) - \frac{C_2 \cdot q}{a \cdot P} \cdot$$

$$\cdot \; (\cos \, a \, l_2 - 1 + a \, l_2 \cdot \sin \, a \, l_2) + \frac{q^2 \cdot l_2^3}{24 \cdot P^2} + \frac{M_2^2}{2 \cdot P^2 l_2} = 44 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\dfrac{P}{E \cdot I}}} \cdot$$

The track sections adjacent to those where the buckling takes place will now be considered. As stated above, the variations in the length of the buckled and non-buckled sections compensate each other (fig. 9).

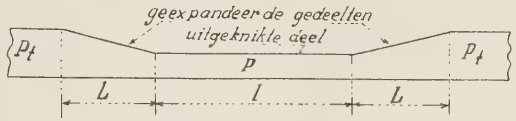


Fig. 9.

Note. — Geexpandeerde gedeelten = expanded sections. — Uitgeknikte deel = buckled section.

In the former the axial force has decreased from P_t to P. The lineal expansion is :

$$\Delta_2 l = \frac{(P_t - P) \cdot l}{E \cdot F} = \frac{(P_t - P) \cdot al}{a \cdot E \cdot F} = \frac{(P_t - P) \cdot al}{E \cdot F \cdot \sqrt{\dfrac{P}{E \cdot I}}} \cdot$$

The total variation in the chord becomes :

$$\Delta_1 l - \Delta_2 l.$$

In any expanded section of the track the difference in length is given by :

$$\Delta L = \int_0^L \frac{(P_t - P) \cdot x}{l} \, dx = \frac{(P_t - P) \cdot L}{2 \cdot E \cdot F}.$$

Representing the resistance offered by longitudinal friction by p, we have :

$$P_t = P + p \cdot L \qquad \text{or} \qquad L = \frac{P_t - P}{p};$$

consequently :

$$\Delta L = \frac{(P_t - P)^2}{2 \cdot p \cdot E \cdot F}.$$

The general equation covering variations in the length is :

$$2 \, \Delta L = \Delta_1 l - \Delta_2 l$$

and becomes for the first example :

$$\frac{(P_t - P)^2}{p \cdot E \cdot F} = 76 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}} - \frac{9 \cdot (P_t - P) \cdot \sqrt{\frac{E \cdot I}{P}}}{E \cdot F}.$$

Resolving this we get :

$$q^2 = \frac{(P_t - P)^2 \cdot P^3 \cdot \sqrt{\frac{P}{E \cdot I}} + 9 \cdot p \cdot P^3 \cdot (P_t - P)}{76 \cdot p \cdot E \cdot I \cdot E \cdot F}.$$

For the second example this same general equation becomes :

$$\Delta L = \Delta_1 l - \Delta_2 l$$

or

$$\frac{(P_t - P)^2}{2 \cdot p \cdot E \cdot F} = 29 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}} - \frac{6.4 \cdot (P_t - P) \cdot \sqrt{\frac{E \cdot I}{P}}}{E \cdot F};$$

and from this we have :

$$q^2 = \frac{(P_t - P)^2 \cdot P^3 \cdot \sqrt{\frac{P}{E \cdot I}} + 12.8 \cdot p \cdot P^3 \cdot (P_t - P)}{58 \cdot p \cdot E \cdot I \cdot E \cdot F}.$$

For the third example, we have :

$$q^2 = \frac{(P_t - P)^2 \cdot P^3 \cdot \sqrt{\frac{P}{E \cdot I}} + 15.8 \cdot p \cdot P^3 \cdot (P_t - P)}{92 \cdot p \cdot E \cdot I \cdot E \cdot F}.$$

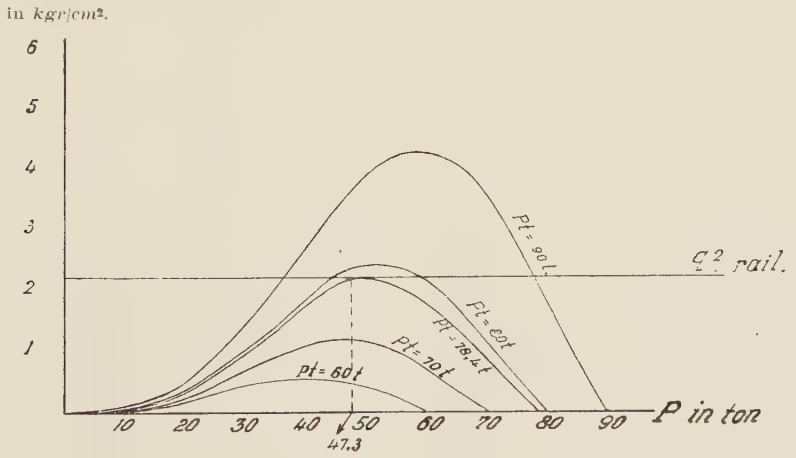
and for the fourth case :

$$q^2 = \frac{(P_t - P)^2 \cdot P^3 \cdot \sqrt{\frac{P}{E \cdot I}} + 19.0 \cdot p \quad P^3 \cdot (P_t - P_t}{110 \cdot p \cdot E \cdot 1 \cdot E \cdot F} -.$$

A common equation applying to all cases generally is :

$$q^2 = \frac{(P_t - P)^2 \cdot P^3 \cdot \sqrt{\frac{P}{E \cdot I}} + \gamma p \cdot P^3 \cdot (P_t - P)}{\delta p \cdot E \cdot I \cdot E \cdot F} = f(P \cdot P_t) \quad . \quad (20)$$

γ and δ being constants depending solely on the assumption of buckling adopted (fig. 10).



Graphical representation of

$$q^2 = \frac{(P_t - P^2)\ P^3 \sqrt{\frac{P}{E.\ I.}} + 9\ p.\ P^3\ (P_t - P)}{76\ p.\ E.\ I.\ E.\ F.}$$ in terms of P.

For $P_t = 90$ tonnes, 80 t., 78.4 t., 70 t. and 60 t.
$p = 3.62$ kgr./cm². $I_v = 2.074$ cm⁴. $F = 85$ cm².
q rail $= 1.49$ kgr./cm². — Size of sleeper : $200 \times 22 \times 12$ cm.

Fig. 10.

Differentiating now q relatively to P we get :

$$\frac{dq}{dP} = \frac{1}{2 \cdot \sqrt{f(P \cdot P_t)}} \cdot \frac{df(P \cdot P_t)}{dP} = 0 \quad . \quad . \quad . \quad . \quad . \quad (21)$$

which assumes that either the denominator is ∞, or the numerator zero. The second condition alone is of practical value, and gives :

$$\frac{df\,(P \cdot P_t)}{dP} = (3\tfrac{1}{2} \cdot P_t - 3\tfrac{1}{2} \cdot P) \cdot (P_t - P) \cdot \sqrt{P} + C \cdot (3 \cdot P_t - 4 \cdot P) = 0 \quad (22)$$

In this equation $C = \gamma \cdot p \cdot \sqrt{E \cdot I}$ is a constant depending on the assumption of buckling and the track construction adopted.

Putting $P = \alpha\, P$, equation (22) can be written :

$$\frac{P_t{}^3}{C^2} = \frac{(3 - 4 \cdot \alpha)^2}{\alpha \cdot (1 - \alpha)^2 \cdot (3\tfrac{1}{2} - 5\tfrac{1}{2} \cdot \alpha)^2} = Y_1$$

or :

$$P_t = \sqrt[3]{Y_1} \cdot \sqrt[3]{C^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

Assuming :

$$\omega = 5 \cdot q^2 \cdot p \cdot E \cdot F \cdot E \cdot I \cdot \sqrt{E \cdot I}$$

we get :

$$\omega = (1 - \alpha^2 \cdot \alpha^3 \cdot \sqrt{\alpha} \cdot (P_t)^{\frac{11}{2}} + C \cdot (1 - \alpha) \cdot \alpha^3 \cdot (P_t)^4.$$

Combining this with (23) we get :

$$\frac{\omega}{\sqrt[3]{C^{11}}} = (1 - \alpha)^2 \cdot \alpha^3 \cdot \sqrt{\alpha} \cdot Y_1{}^{\frac{11}{6}} + (1 - \alpha) \cdot \alpha^3 \cdot Y_1{}^{\frac{8}{6}} = Y_2 \quad . \quad . \quad (24)$$

We have succeeded in isolating in the first member of the equation nothing but the factors depending on the construction of the track and the particular case of buckling under consideration, while the second is only a function of α.

In applying the above-mentioned method of calculation, we get the smallest possible value for P_t, but generally find a fairly large ordinate of deflection. Assuming a certain value for the latter, the corresponding value of P can be arrived at by the above equations, and P_t can thus be drawn as varying with the deflection f.

It may now be asked, what value can the buckling load reach in a track which buckles under a large number of waves? In such cases the equation for the compensation arising from the changes in length becomes, the number of waves being n :

$$2 \cdot \Delta L = n \cdot \Delta_1 l - n \cdot \Delta_2 l.$$

Consequently (20) becomes :

$$q^2 \quad \frac{(P_{t_1} - P)^2 \cdot P^3 \cdot \sqrt{\frac{P}{E \cdot I}} + \frac{n}{2} \cdot \gamma \cdot P^3 \cdot p \cdot (P_t - P)}{\frac{n}{2} \cdot 5 \cdot p \cdot E \cdot I \cdot E \cdot F}$$

n becoming infinitely large, the first term of the numerator is negligible with respect to the next, and therefore :

$$q^2 = \frac{\gamma}{\delta} \cdot \frac{P^3 \cdot (P_t - P)}{E \cdot I \cdot E \cdot F}.$$

Furthermore :

$$Y_2 = \frac{\omega}{\sqrt[3]{C^{11}}} = \frac{\frac{n}{2} \cdot \delta \cdot q^2 \cdot p \cdot E \cdot F \cdot E \cdot I \cdot \sqrt{E \cdot I}}{\left(\frac{n}{2}\right)^{\frac{11}{3}} \cdot (\gamma \cdot p \cdot \sqrt{E \cdot I})^{\frac{11}{3}}} = 0 \text{ for } n = \infty.$$

$$Y_2 = 0 \text{ for } y_1 = 0, \text{ this being for } 3 - 4\alpha = 0 \text{ or } \alpha = \frac{3}{4}$$

or :
$$P = \frac{3}{4} P_t$$

and consequently :
$$q^2 = \frac{\gamma}{\delta} \cdot \frac{1}{4} \cdot \left(\frac{3}{4}\right)^3 \cdot \frac{P_t^4}{E \cdot I \cdot E \cdot F}$$

or
$$P_t = \sqrt[4]{\frac{256}{27} \cdot \frac{\delta}{\gamma}} \cdot \sqrt{q} \cdot \sqrt[4]{E \cdot I \cdot E \cdot F}.$$

For the first example, with vertical deformation, we get :

$$P_t = 2.99 \cdot \sqrt{q} \cdot \sqrt{E \cdot I \cdot E \cdot F}.$$

For deformation in the horizontal plane of the track, the case must be considered in which the two moments M_1 and M_2 are nil. Then we have :

$$al = 4.5 \text{ and } \Delta_1 l = 13.8 \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}} \text{ and } q = 0.196 \cdot \frac{P^2 \cdot f}{E \cdot I}$$

as well as
$$P_t = 2.32 \sqrt{q} \cdot \sqrt[4]{E \cdot I \cdot E \cdot F}.$$

These values differ slightly from those calculated by Professor de Vries-Broekman (Cf. « Knikvastheid van een naadloos spoor », *De Ingenieur*, May 6th 1932, No. 19). In this paper $\Delta_1 l$ is deduced from the equation :

$$\Delta_1 l = \frac{\pi^2}{4} \cdot \frac{f^2}{l}$$

For vertical buckling, for instance, Prof. de Vries-Broekman has found :

$$q = \alpha \cdot \frac{\pi^2}{l^2} \cdot \left(P - \frac{4 \cdot \pi^2 \cdot E \cdot I}{l^2}\right) \cdot f \text{ and } \alpha = 1.05.$$

and also :

$$P = \frac{3}{4} P_l \quad \text{while} \quad P_l = 4 \cdot \frac{4 \cdot \pi^2 \cdot E \cdot I}{l^2}.$$

Therefore :

$$\frac{4 \cdot \pi^2 \cdot E \cdot I}{l^2} - \frac{P_l}{4} = \frac{1}{4} \cdot \frac{4}{3} P = \frac{1}{3} P$$

whence we get :

$$q = \frac{1.05 \cdot \pi^2}{l^2} \cdot \frac{2}{3} \cdot P \cdot f = \frac{1.05 \cdot 2 \cdot \pi^2}{3 a^2 l^2} \cdot \frac{P^2 \cdot l}{E \cdot I}.$$

And then :

$$P = \frac{3}{4} P_l = \frac{3 \cdot 4 \cdot \pi^2 \cdot E \cdot I}{l^2}, \text{ consequently } a^2 l^2 = \frac{P \cdot l^2}{E \cdot I} = 12 \cdot \pi^2 \text{ or } al = \sim 11.$$

Therefore :

$$q = \frac{2.1}{36} \cdot \frac{P^2 \cdot f}{E \cdot I} = 0.058 \cdot \frac{P^2 \cdot f}{E \cdot I}.$$

Above we found that : $al = 9$ and $q = 0.064 \cdot \frac{P^2 \cdot f}{E \cdot I}$.

Consequently :

$$f = \frac{q \cdot E \cdot I}{0.058 \cdot P^2}$$

whence we have :

$$\Delta_1 l = \frac{\pi^2}{4} x \frac{q^2 \cdot E^2 \cdot I^2}{l \cdot 0.058^2 \cdot P^4} = \frac{\pi^2}{4 \cdot 0.058^2 \cdot a l} \cdot \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}}$$

or

$$\Delta_1 l = \sim 68 \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}}.$$

In the same way we found :

$$\Delta_1 l = 76 \frac{q^2 \cdot E \cdot I}{P^3 \cdot \sqrt{\frac{P}{E \cdot I}}}.$$

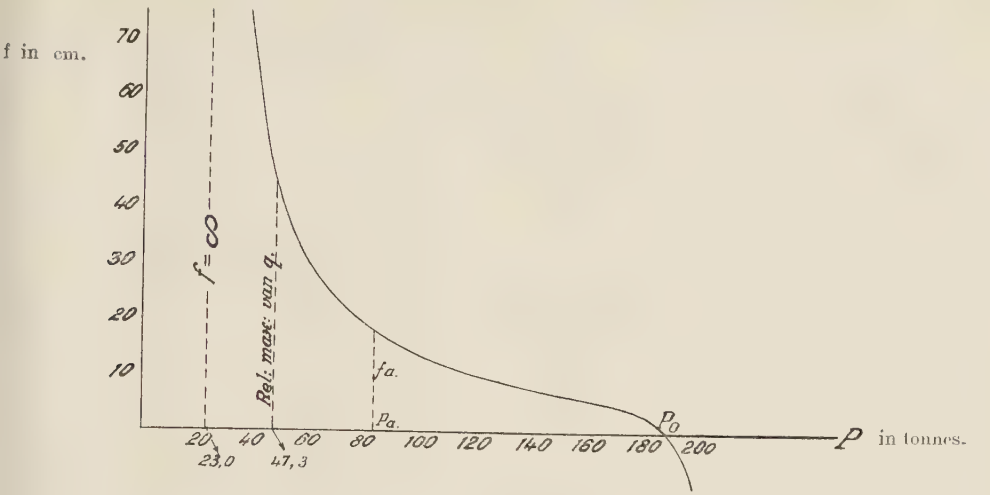
Thus we get

$$\sqrt[4]{\frac{256\,\delta}{27\,\gamma}} = \sqrt[4]{\frac{256 \times 68}{27 \times 11}} = \sqrt[4]{58.6127} = 2.76$$

which justifies the differences in value obtained for the constant factors.

Let us now consider the condition of equilibrium of the buckled section of the track. Let us consider the first example and draw the function :

$$f = \frac{q \cdot l}{2 \cdot a \cdot P} \cdot \left(\frac{al}{4} - \tan \frac{al}{4} \right).$$



Buckling load according to EULER.

Graphical representation of :

$$f = \frac{q^e}{2aP}\left(\frac{al}{4} - \tan.\frac{al}{4}\right) \text{ in terms of P.}$$

$I_v = 2\ 074\ cm^4$. $F = 85\ cm^2$. $l = 27.31$ m.

q rail $= 1.49$ kgr.cm^1 — Dimensions of sleeper : $200 \times 22 \times 12$ cm.

Fig. 11.

We get a curve of periodic variation, he first practical significance of which s represented in figure 11. We see rom it that a certain value of P_a corresponds to a certain value of f_a. If f iminishes, P increases; the ends of the uckled section not being fixed, it traightens itself if P remains constant. s P remains constant, and f increases, the track buckles completely and hence equilibrium is unstable. Actually total buckling is limited by the adjacent sections of track which localise the ends of the buckled part. At P_o the ordinate of curvature becomes nil, but then the curve of elasticity is formed of two branches, and we have :

$$y = \frac{-q \cdot l_2}{2 \cdot P \cdot}\left[\frac{\cos ax - 1}{al \cdot \tan \frac{1}{2} al} + \frac{\sin ax}{al}\right] + \frac{\frac{1}{2} q \cdot x (l - x)}{P}.$$

We see that y is nil if $P = \backsim$, i.e. $al = \sim$, because the factor al tan $\frac{1}{2} al$ can be $\backsim$ nil if $al = \sim$. But the alue of this factor can vary between $-\sim$ and $-\sim$ *which allows us to conude that theoretically an infinitely lare axial force alone can upset the equibrium of an ideally lineal track, but that in this case equilibrium is very unstable, a small disturbing influence sufficing to distort such a system.* Actually,

Fig. 12. — Going over from 2 waves to 3.

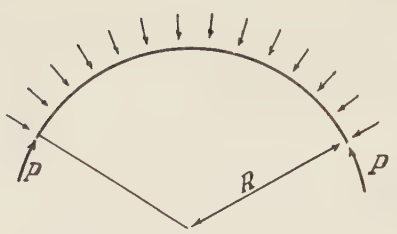


Fig. 13.

an ideal track is nothing but an illusion, and in practice various factors tend to cause buckling with a finite axial force as, for instance, in the case of a small initial curvature of the track.

Fig. 14.

In the case of deformation in the horizontal plane, other factors mentioned above come into play, such as frictional resistance of the sleepers, the large variation in the moment of inertia, and defective fastenings; as pointed out above, the resistance offered by the sleepers is at first in some measure proportional to the displacement. Horizontal deformation is possible from finite axial forces. In such a case the buckling load is given approximately by ENGESSER's formula :

$$P_{t\ (\text{elastic})} = 2\sqrt{\frac{E \cdot I \cdot A'}{c}}$$

by representing :

(1) the resistance of the sleepers per unit length of displacement by A',

(2) the sleeper spacing by c,

and then making $I = 2I_y$ (I_y = the moment of inertia of a single rail).

In figures 16, 17 and 18 we show the conditions applying to track laid with No. 3 type rails on the Dutch East Indies State Railways.

As can be seen, the value of P_t (elastic) is very great. Having regard also to other disturbing influences, *we conclude that the axial force P_t must first become much larger than $P_{t\,\min}$, deduced above, before the track can buckle dangerously. Consequently an axial force a little less than $P_{t\,\min}$ can already be tolerated.* We may also point out that the track can become deformed in any number of waves, but that a deformation with two waves gives the minimum buckling load in the horizontal plane. The most unfavourable vertical deformation is with a single wave.

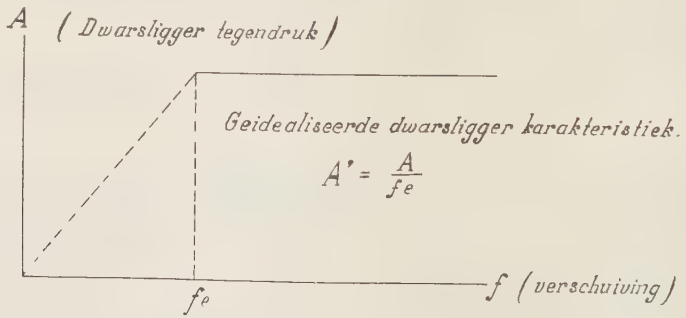
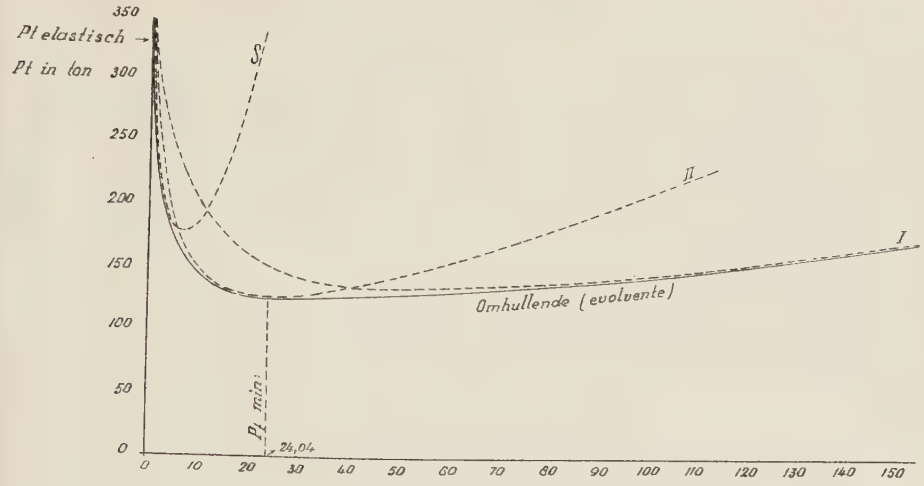


Fig. 15.

Note. — Dwarsligger tegendruk = reaction of sleeper. — Geidealiseerde dwarsligger karakteristiek = characteristics of an ideal sleeper. — Verschuiving = displacement.



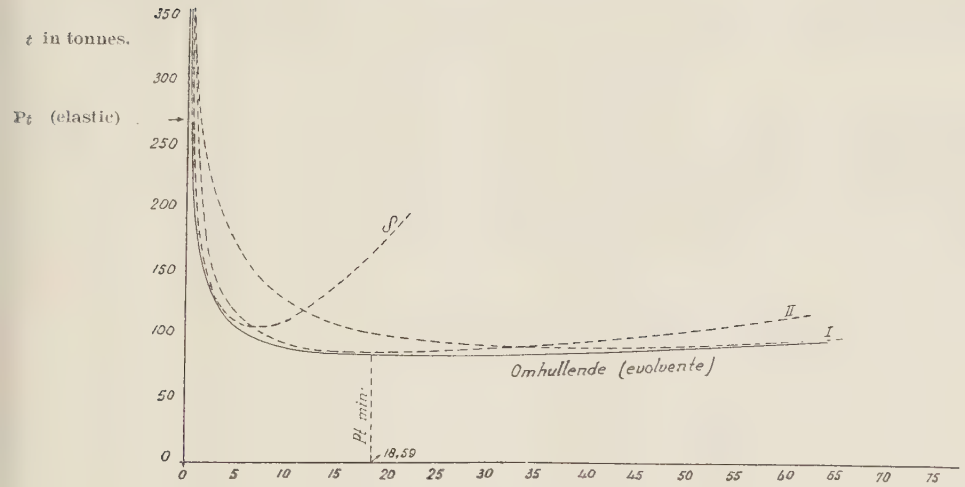
P_t in terms of the mean deflection for examples I, II and ∞.

Wooden sleepers and broken stone ballast

$I_h = 1\,500$ cm⁴. $F = 85$ cm². $p = \frac{250}{69} = 3.62$ kgr./cm¹ $q = \frac{550}{69} = 7.97$ kgr./cm¹.

Size of sleepers : 200×22×12 cm. Sleeper spacing : 69 cm.

Fig. 16. — Horizontal buckling.



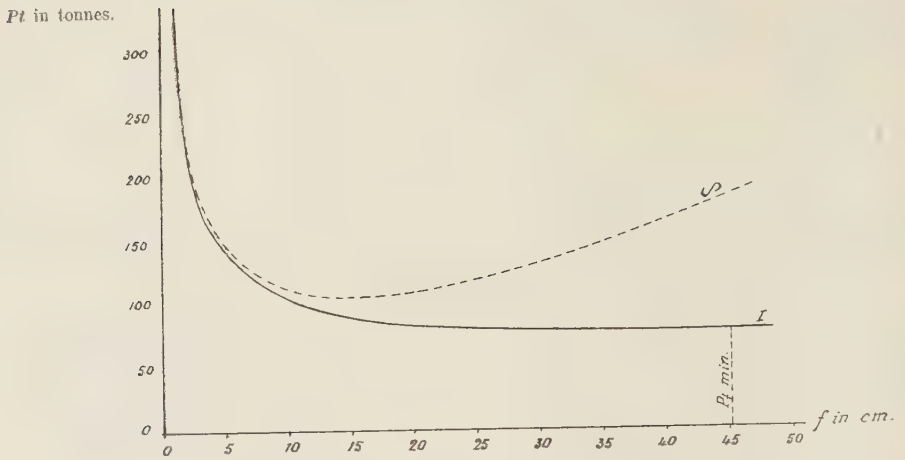
P_t in terms of the mean deflection f for exemples I, II and ∞.

Wooden sleepers with sand ballast.

$I_h = 1\,500$ cm⁴, $F = 85$ cm². $p = \frac{250}{69} = 3.62$ kgr./cm, $q = \frac{200}{69} = 2.90$ kgr./cm¹.

Size of sleepers : 200 × 22 × 12 cm. — Sleeper spacing : 69 cm.

Fig. 17. — Horizontal buckling.



P_t in terms of the mean deflection f for examples I and S.

Iy = 2 074 cm⁴. F = 85 cm². $p = \frac{250}{69} = 3.62$ kgr./cm¹ $q = 1.49$ kgr./cm¹.

Size of sleepers : 200 × 22 × 12 cm. Sleeper spacing : 69 cm.

Fig. 18. — Vertical buckling.

Our final conclusion is that maintenance of welded track should in the first place be directed to keeping it in perfect alignment, and the fastenings perfectly tight.

Exemple of calculations.

The weight of the No. 3 type permanent way is 1.49 kgr./cm¹, the sleepers being spaced at 69 cm.

The transverse resistance of the sleepers in the surrounding sand ballast is $\frac{200}{69} = 2.90$ kgr./cm¹ while the longitudinal resistance comes to $\frac{250}{69} = 3.62$ kgr./cm¹.

Then F = 85 cm², $I_x = 2074$ cm⁴, $I_y = 1500$ cm⁴.

For vertical buckling : $\gamma = 9$ and $\delta = 76$, we get :

$$c^2 = 4623; \quad \frac{\omega}{\sqrt[3]{C^{11}}} = 189; \quad P_{t\,min} = 78.4 \text{ tonnes}; \quad P = 47.3 \text{ tonnes}.$$

For horizontal buckling : First exemple, $\gamma = 9$; $\delta = 76$:

$$c^2 = 3344; \quad \frac{\omega}{\sqrt[3]{C^{11}}} = 799; \quad P_{t\,min} = 93.6 \text{ tonnes}; \quad P = 57.7 \text{ tonnes}.$$

Second example : $\gamma = 12.8$; $\delta = 58$:

$$c^2 = 6764; \quad \frac{\omega}{\sqrt[3]{C^{11}}} = 168; \quad P_{t\,min} = 87.3 \text{ tonnes}; \quad P = 57.6 \text{ tonnes}.$$

Table for $\sqrt[3]{\gamma_1}$ et γ_2 in terms of α.

	$\sqrt[3]{\gamma_1}$	γ_2	α	$\sqrt[3]{\gamma_1}$	γ_2	α	$\sqrt[3]{\gamma_1}$	γ_2	α	$\sqrt[3]{\gamma_1}$	γ_2	α	$\sqrt[3]{\gamma_1}$	γ_2
	4.22	0.00058	0.570	3.33	29.24	0.597	4.35	117.85	0.626	9.58	7543.16			
	2.56	0.00944	0.571	3.35	30.19	0.598	4.41	126.62	0.627	10.17	9516.95	0.662	4.35	123.40
	2.12	0.035	0.572	3.37	31.24	0.599	4.47	136.13	0.628	10.98	15682.37	0.663	4.23	106.53
	1.87	0.15	0.573	3.40	32.71	0.600	4.54	147.16	0.629	11.87	23906.67	0.664	4.11	92.46
	1.86	0.45	0.574	3.43	33.93	0.601	4.61	159.27	0.630	12.80	35791.84	0.665	3.93	73.79
	2.00	1.29	0.575	3.46	35.74	0.602	4.68	172.65	...	...	...	0.666	3.84	65 13
	2.42	5.02	0.576	3.49	37.22	0.603	4.76	188.24	...	...	...	0.667	3.74	57.67
50	2.94	15.09	0.577	3.52	38.83	0.604	4.84	205.73	0.640	18.03	230251.07	0.668	3.66	51.29
51	2.96	15.48	0.578	3.55	40.59	0.605	4.93	225.02	0.641	15.21	91588.71	0.669	3.53	42.66
52	2 97	15.90	0.579	3.58	42.54	0 606	5.02	248.16	0.642	13.55	48897.89	0 670	3.45	38.24
53	2.99	16.46	0.580	3.61	44.67	0.607	5 12	274.07	0.643	12.00	25385.15	0.671	3.37	34.35
54	3.01	16.94	0.581	3.64	46.55	0.608	5.22	303.28	0.644	10.70	13668.12	0.672	3.27	29.18
55	3.02	17.45	0.582	3.68	46.75	0.609	5.32	336.32	0.645	9.81	8595.65	0.673	3.20	26.45
56	3.04	18.00	0.583	3.71	51.34	0.610	5.44	376.36	0.646	9.14	5887.84	0.674	3.10	22.71
57	3.06	18.59	0.584	3.75	53.90	0.611	5.56	422.06	0.647	8.52	4071.69	0.675	3.04	20.65
58	3.07	19.07	0.585	3.79	56.82	0.612	5.70	479.14	0 648	7.97	2848.63	0.676	2.95	17.91
59	3.09	19.74	0.586	3.82	59.82	0.613	5.82	539.51	0.649	7.52	2093 67	0.677	2.90	16.37
60	3.11	20.46	0 587	3.86	63.20	0 614	5.98	622.27	0.650	7.09	1530.16	0.678	2.82	14.29
61	3.13	21.05	0.588	3 90	66.79	0 615	6.14	711.39	0.651	6.71	1146.61	0.679	2.77	13.11
62	3.15	21.89	0.589	3.94	70.65	0.616	6.32	826.59	0.652	6.43	915.09	0 680	2.70	11.53
63	3.17	22.58	0.590	3.99	74.87	0.617	6.50	962.01	0.653	6.11	704.46	0.690	2.15	3.93
64	3.19	23.33	0.591	4.04	79.57	0.618	6.72	1142.49	0.654	5.84	557.78	0.7	1.73	1.46
65	3.21	24.13	0.592	4.08	84.41	0.619	6.94	1362.74	0.655	5.60	450.16	0.8	1.15	0.22
66	3.24	24.99	0.593	4.13	89.85	0.620	7.18	1628.08	0.656	5.38	362.76	0.9	2.08	5.24
67	3.26	25.93	0.594	4.18	96.08	0.621	7.47	1998.79	0.657	5.17	297.28	0.95	4.48	25.36
68	3.28	26.94	0.595	4.23	102.46	0.622	7.81	2541.75	0.658	4.99	247.11	0.96	5.24	39.31
69	3.31	28.05	0.596	4.29	109.48	0.623	8.14	3147.79	0.659	4.80	202.06	0.97	6.44	69.89
						0.624	8.52	4049.72	0.660	4.64	169.74	0.98	10 65	327.27
						0.625	9.03	5490.47	0.661	4.49	144.06	0.99	13.21	436.85

Third example : $\gamma = 15.8$; $\delta = 92$:

$$c^2 = 10304\,; \quad \frac{\omega}{\sqrt[3]{C^{11}}} = 123\,; \quad P_{t\,\min} = 94.7 \text{ tonnes}\,; \quad P = 62.7 \text{ tonnes}.$$

Fourth example : $\gamma = 19.0$; $\delta = 110$:

$$c^2 = 14901\,; \quad \frac{\omega}{\sqrt[3]{C^{11}}} = 75.0\,; \quad P_{t\,\min} = 97.0 \text{ tonnes}\,; \quad P = 64.5 \text{ tonnes}.$$

For vertical buckling : $P_t = 108.4$ tonnes (number of waves $= \sim$),
and *for horizonlal buckling* : $P_t = 108.9$ tonnes (number of waves $= \sim$).

For deformation in the horizontal plane we get, therefore :

1 wave,	$P_{t\,\min} =$	93.6 tonnes,	
2 waves,	$P_{t\,\min} =$	87.3 »	
3 waves,	$P_{t\,\min} =$	94.7 »	
4 waves,	$P_{t\,\min} =$	97.0 »	(See figs. 16, 17 and 18)
$\sim$ waves,	$P_{t\,\min} =$	108.9 »	

CHAPTER V.

Stability of curved track.

Let us assume a section of curved track, the sleepers of which offer no resistance to slipping sideways. If the temperature rises, the track will bulge towards the outside of the curve, lengthening in proportion to the rise in temperature. The track will not be subjected to any compressive force. If, on the other hand, the sleepers offer resistance to slipping sideways, the track will remain in place so long as the compressive force is equalised by transverse forces due to sleeper frictional resistance.

If the maximum frictional force is taken as A,

the sleeper spacing as c,

the radius of the curve as R,

the compressive force can attain the value $P_t = \frac{AR}{c}$ = without any perceptible deformation occurring, even if the rigidity of the type of track equipment used is small.

As soon as the compressive force exceeds P_t, deformation of the track becomes possible.

Assuming that the ballast is perfectly homogeneous, the curved track will move outwards so as to expand without buckling. But perfect homogeneity is an illusion and as soon as $P_t = \frac{AR}{c}$ is exceeded, buckling becomes possible.

Curves being generally very long, in any case longer than the length of buckling corresponding to $P_t = \frac{AR}{c}$, this force constitutes the buckling load.

Messrs. AMMANN and GRUENEWALDT have dealt, in their publications, with a case of buckling as shown in fig. 14 (see *Organ für die Fortschritte des Ei-*

senbahnwesens, No. 4, 15th February, *1931),* but this example has no practical significance. The deformation should be dealt with as in example No. 2 above.

We venture to refer to two anomalies in the article by Professor F. CORINI « Track laying for high speeds » (See *Bulletin of the International Railway Congress Association,* April, 1936). On page 384, we read :

« When $\frac{a\,L}{2} = (2\,n + 1)\,\frac{\pi}{2}$, we have : $\cos\frac{a\,L}{2} = 0$ and $f_1 = \infty$.

To prevent the profile of the line being dangerous, $aL < \pi$.

Formula 18 on the same page should be interpreted as follows :

When the axial force F_1 has the value $F_1 = \frac{\pi^2\,EI}{l^2}$ (which is the case if $aL = \pi$), only an infinitely great ordinate of curve can insure stability in such a system.

It is impossible for a system subjected to a buckling load and an equivalent transverse load to have such an ordinate of curvature *in a direction opposed to the latter load,* because the equilibrium of a girder subject to a buckling load only is indifferent, and consequently when a transverse load is added to it, it has a tendency to re-straighten itself.

A second anomaly must be pointed out on page 386 of the same article : the author thinks that if the ordinate of curvature $f_1 = 0$, the curve of elasticity becomes a straight line, but it has been shown above that this is not generally so.

A third anomaly has been corrected by Mr. BATICLE *(Bulletin of the Railway Congress,* September 1936, p. 985).

Recapitulation.

With the data at our disposal we can calculate

$$c = \gamma \cdot p \cdot \sqrt{E \cdot I} \text{ and } \omega = \delta \cdot q^2 \cdot p \cdot E \cdot F \cdot E \cdot I \cdot \sqrt{E \cdot J}$$

and hence

$$y_2 = \frac{\omega}{\sqrt[3]{C^{11}}} - \frac{\omega}{C^{\frac{11}{3}}}$$

The table gives direct corresponding values for α and $\sqrt[3]{y_1}$. For the latter expression we use the smallest value found, whence

$$P_t = \sqrt[3]{Y_1} \cdot \sqrt[3]{c^2} \quad \text{and} \quad P = \alpha\,P_t.$$

We take :

for a single wave :	$\gamma = 9$	and $\delta = 76.$
for two waves :	$\gamma = 12.8$	and $\delta = 68.$
for three waves :	$\gamma = 15.8$	and $\delta = 92.$
for four waves :	$\gamma = 19.0$	and $\delta = 110.$

The number of waves being ∞ we get :

(a) for vertical buckling :

$$P_t = 2.99 \sqrt{q \cdot E} \cdot \sqrt[4]{I \cdot F}$$
$$P = 0.75\ P_t$$

(b) for horizontal buckling :

$$P_t = 2.32 \cdot \sqrt{p \cdot E} \cdot \sqrt[4]{I \cdot F}$$
$$P = 0.75\ P_t$$

when :

the longitudinal frictional resistance per unit length = p;

the transverse frictional resistance per unit length = q;

the moment of inertia = I;

the cross section = F.

For curved track : $P_t = \frac{A\,R}{c}$

when :

the maximum transverse frictional resistance of the sleepers = A;

the sleeper spacing = c;

and the radius of the curve = R.

* * *

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(b) « Versuche über die Wirkung von Längskräften im Gleis » (Experiments on the action of longitudinal forces in the permanent way),

I : No. 16, 15th August 1928;
II : No. 14, 15th July 1929;
III : No. 22, 15th November 1929;
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(b) « Versuche der Königliche Ungarischen Staatsbahnen über die Standsicherheit des Gleises » (Experiments of the Hungarian State Railways on the stability of the permanent way) ; No. 6, 15th March 1932;

(c) « Die Ausdehnung der Schienen durch die Wärme » (Expansion of rails due to heat) ; No. 16 of the 15th August 1928.

[621. 133.1]

The combustion of liquid fuel (mazout) in locomotive boilers,

by Basile CERNAT,
Chief Engineer of the Rumanian State Railways.

The use of the liquid fuels, mazout and tar, for firing locomotive fireboxes, as well as for stationary boilers, has been very much extended during the last ten years, on account of the special technical advantages that they afford, as compared with solid fuels. This applies particularly to those countries where these oil fuels are cheap, and Rumania is a notable example of this, as she possesses extensive oil-fields where mazout can be had at a very low price.

It is well known that, in order to obtain proper results from the combustion of these liquid fuels, they must be both well atomised and mixed with air.

When the first trials were carried out, from 1870 to 1880, the results were not satisfactory, although petroleum itself, which is more vaporisable than mazout, was used, because it was burnt in pans placed on the grate in the firebox, or was converted into gas in a coiled tube fitted in the firebox.

The fuel was first atomised in 1880, and thenceforward better results were obtained.

In Rumania, mazout was first used for firing locomotives in 1887.

Atomisation of the liquid fuel.

There are several methods for obtaining proper atomisation; let us quote the following methods :

(1) *Atomisation by steam or compressed air;*

(2) *Atomisation by compression of the fuel,* at a pressure of 2 to 15 atm. (28.4 to 213.3 lb. per sq. in.), in which process the fuel is forced through a very restricted aperture;

(3) *Atomisation by various mechanical processes.* A rotary motion is imparted to the liquid, which becomes atomised by the effect of centrifugal force.

Mention may be made of yet another interesting method of atomisation, of the same kind as the internal combustion engine carburetter, but this has not yet been perfected.

The method of atomising the mazout by compressed air is only employed where no steam is available; and this is the case as regards hot water caloriferes, as well as marine boilers, for which there is no fresh water available.

Atomisation by compression and mechanical atomisation have begun to be developed recently on a large scale.

But it must be borne in mind that these systems are liable to get out of order rather frequently, and can therefore only be utilised in those instances where such troubles will not seriously hamper the working.

Everything therefore points to atomisation by steam as the method to be adopted for industrial boilers, and particularly for locomotives, where reliability in service constitutes an essential feature.

Automatic devices.

Until recently, these devices were called « *injectors* » by reason of the process of atomisation employed, resembling that used for injecting water into boilers. But the name which really fits them is « *mazout atomisers* » or « *mazout burners* », particularly as the high velocity injection of the mazout constitutes a drawback for the firebox walls which receive this injection, and it would not be logical to link the name of the device with an action which might suggest the idea of a shortcoming.

This article will only deal with steam atomisers for locomotives.

As a results of the development in the use of mazout for firing boilers, very many atomiser improvements have been brought out. It may be said that nowadays any mechanician of an inventive turn of mind has his own atomiser of empirical construction, although this has not prevented serious designers from studying the question scientifically.

The *Dragu* and *Cosmovici* atomisers on the locomotives of the Rumanian Railways date back more than 30 years, and it was felt that a new system, of improved design, would enable appreciable savings of fuel to be made, and at the same time would preserve simplicity of operation and reliability in service.

A few isolated trials confirmed this forecast and led the Rumanian Railway Administration to organise a competition, with a view to the adoption of the best system.

After a few devices which did not give satisfactory results had been eliminated, the following six types were retained for trial purposes : *Jelinek*, *Popescu-Lemaître*, *Horgos*, *Molnar*, *Marsi*, and *Serbescu-Grecu*.

Description of the devices.

Before dealing with the trials and their results, we shall describe these devices and how they work; but we shall only concern ourselves with the arrangement of the burner end of the atomiser, and ignore the body of the device which is of the usual construction and of no special interest.

As the new devices follow the lines of the old *Dragu* and *Cosmovici* atomisers, we shall first of all describe these old atomisers.

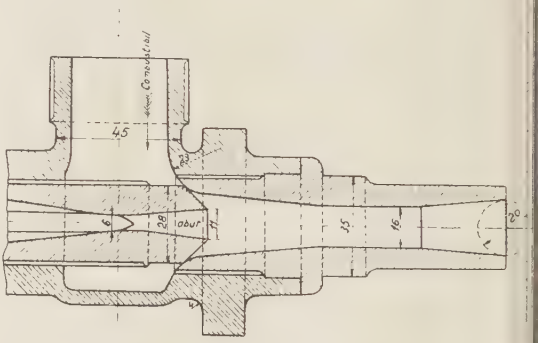

Fig. 1. — Dragu atomiser.

(I) *The Dragu atomiser* (fig. 1) is of the round type producing a conical flame. The steam, which enters through the centre line of the apparatus, passes into a nozzle which produces a convergent and divergent action in such a manner as to impart a high velocity to it, and ensures the proper mixing of the steam with the mazout, which latter is sucked into this nozzle.

The very long flame is suitable for long fireboxes.

The atomisation obtained is efficient, but the apparatus makes a noise which is trying to the staff.

(II) *The Cosmovici atomiser* (fig. 2)

is of the flat type, and produces a sheet of flame in the firebox.

The steam is supplied through an orifice of rectangular shape, 100 mm. × 1 mm. (3.94 × 0.0394 in.) in size.

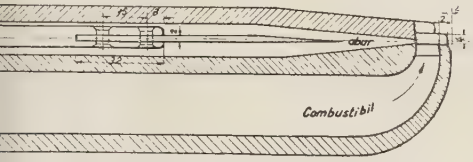

Fig. 2. — Cosmovici atomiser.

The steam breaks up the mazout into particles as the mazout emerges below the jet of steam, and projects it into the firebox.

The atomisation is not quite perfect, but the flame is distributed inside the firebox in a very satisfactory manner. The mixture with the air inside the firebox is very much better than with the apparatus previously described, especially for prismatic fireboxes.

(III) *The Popescu-Lemaître atomiser* (fig. 3).

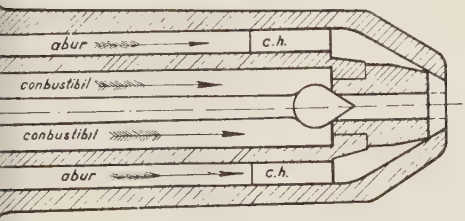

Fig. 3. — Popescu-Lemaître atomiser.

We have seen that, in the case of the Dragu atomiser, the steam directly enters a nozzle where the mixture of the steam with the mazout is effected. Modern designers have had the idea of imparting a spiral motion to the steam, in order to procure not only more efficient atomisation, but at the same time, a shorter flame.

The *Popescu* atomiser is formed of a central cylindrical body, through which the mazout flows, the delivery of which can be regulated by means of a rod which can even be used to plug up the delivery outlet for the mazout.

This rod can also be used for cleaning the tube.

The central cylindrical body is fitted into a tube of much larger diameter, leaving between them a free passage, through which the steam passes.

In order that the steam in passing along may acquire a spiral motion, spiral blades are arranged in the central body, which produce this motion.

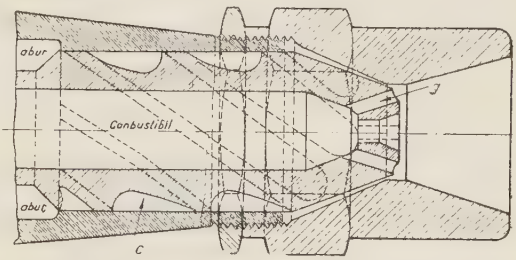

Fig. 4. — Jelinek atomiser.

(IV) *The Jelinek atomiser* (fig. 4) is of similar construction to the preceding one, but has no central rod.

The outlet for the mazout consists of several orifices *(g)*, and the mazout is sucked along by the steam which takes on a spiral motion as the results of travelling through the spiral passage *(c)*.

(V) *The Horgos atomiser.* — There are no marked differences between the Horgos and the Jelinek atomiser.

(VI) *The Marsi atomiser* (fig. 5). — The outlet for the mazout is through several orifices *(s)* of small diameter, and the steam assumes a spiral motion by

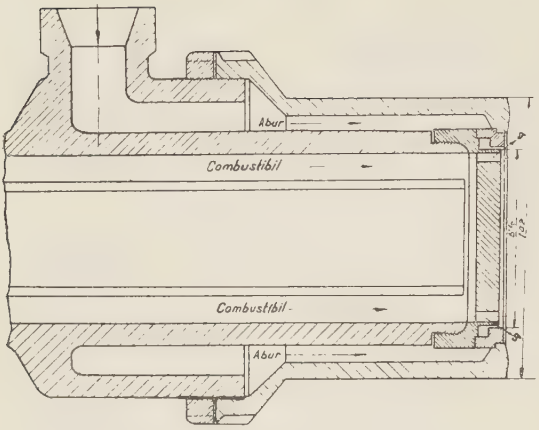

Fig. 5. — Marsi atomiser.

passing through a ring in which there are about 100 spiral grooves.

(VII) *The Molnar atomiser* (fig. 6) is of the flat type. The mazout emerges through a vertical slit, and is sucked

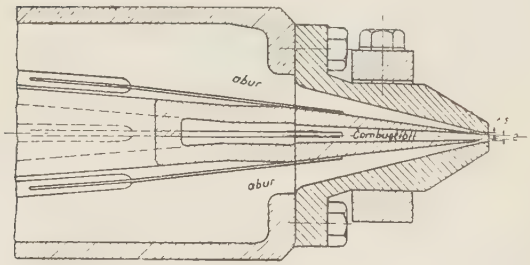

Fig. 6. — Molnar atomiser.

along by the steam which is supplied through two slits placed on both sides of the mazout delivery outlet.

(VIII) *The Serbescu-Grecu atomiser* (fig. 7) is formed of 3 round atomisers fitted in one body. The mazout is fed through 3 divergent pipes. The steam is compelled to assume a spiral motion by passing through several oblique orifices *(g)* arranged round each of these three pipes. The steam attains a very high velocity in passing through the nozzles arranged as a continuation of the holes.

This atomiser must be classed amongst the multiple-flame atomisers, because it gives 3 distinct flames.

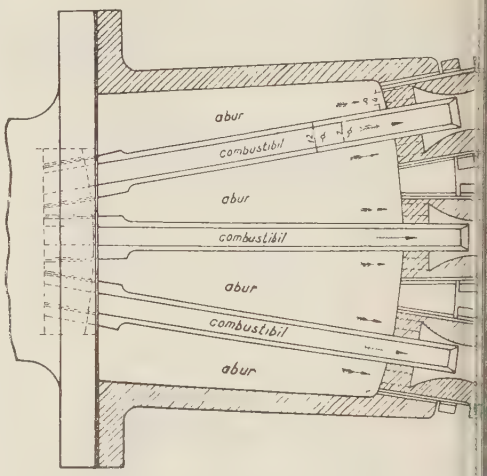

Fig. 7. — Serbescu-Grecu atomiser.

Remarks. — An efficient atomiser must satisfy the following conditions :

(1) *be of simple construction;*

(2) *be very reliable in service;*

(3) *give the necessary delivery for the maximum power of the locomotive;*

(4) *possess a satisfactory efficiency.*

We shall now consider each of the different atomisers previously described from these various points of view.

(1) The *Popescu* atomiser is of simple construction. The mazout is fed through a passage of rather large diameter, fitted with a rod for cleaning purposes, which ensures that the mazout delivery passage cannot become chocked up.

During the trials, it only gave a maximum delivery of mazout of 600 kg (1 322 lb.) per hour. The efficiency of this apparatus can be appreciated from

the results set out in the table of trials given below.

(2) The *Jelinek* and *Horgos* atomisers are also of simple construction, but they are liable to become clogged up, as the mazout passes through holes of rather small diameter.

(3) The *Marsi* atomiser also delivers the mazout through small holes, which can easily become choked up.

Its delivery is also limited to 600 kgr. (1 322 lb.) of mazout per hour.

(4) The *Molnar* atomiser originally gave an insufficient delivery. Mr. Molnar then designed another apparatus of the same kind, with an adequate delivery, having several slits for the passage of the mazout. As this apparatus had no knife-edge for cleaning the slits, it ran the risk of becoming choked up.

(5) The *Serbescu-Grecu* atomiser solved the condition that the Engineer, Mr. Cosmovici, had set out to solve, but without succeeding completely, namely : *his proposal was to achieve a flat atomised jet formed of several flames (4 to 6), separated one from the other, so as to enable the air to surround each flame, and thus ensure an efficient mixing of the air with the mazout.*

This principle is applied to stationary boilers, in which 10 to 20 small atomisers are fitted in a single firebox.

The outlet for the mazout in the *Cosmovici* atomiser is from several distinct pipes, but the jet of steam which is divided up by stops placed in the steam outlet combines together again after it emerges and forms one single flat jet.

The *Serbescu-Grecu* atomiser solves this problem by having 3 distinct atomiser burner ends instead of 4 to 6.

Moreover, the spiral and rotary motion imparted to the steam produces whirling streams which ensure the air being properly mixed with the mazout.

The construction of this apparatus is fairly simple, and as the mazout delivery pipes are of large diameter, there is no fear of their becoming choked up. Nevertheless, as an extra precaution, the apparatus is fitted with a hole at the back through which these pipes can be cleaned out.

The delivery given by this apparatus is only limited by the quantity of mazout which can be burnt in the firebox.

Atomiser trials.

The trials of all the atomisers were carried out under identical conditions, namely : The same locomotive hauling the same load, at the same speed on the same line and using similar fuel.

The comparison between the results obtained was based on the efficiency of the boiler, or — which amounts to the same thing — the evaporating capacity of one kilogramme of fuel obtained with each apparatus.

This capacity was calculated in kgr. of normal steam containing 639 calories (2 560 B.T.U.), in order to take into account the fact that the superheat temperature varied from one atomiser to another.

We did not base the comparison of the fuel consumption figures on the tonne-kilometre, because some very variable factors come into play here, although the loading was the same, viz. : the atmospheric conditions, the fluctuations in the number of passengers, etc.

Precautions were taken to ensure that the conditions of working were the same, viz. : The locomotive boiler was washed out before each atomiser trial, the working pressure maintained, the water fed

in at the same intervals, the grate kept properly covered with coal, etc.

Although every precaution was taken to ensure the locomotives worked in the same manner every time, we encountered some variable conditions produced by unexpected brake applications.

The efficiency was influenced by the steam demand from the boiler, and we have allowed for this.

The production of steam was measured through the quantity of water introduced into the boiler, which water was itself measured by using a tender marked off in graduations. Similarly, the boiler of the locomotive was gauged, in order to ascertain the quantity of the water consumed it contained.

We also took into account the inclination of the tender and of the locomotive, by means of water gauges.

For the purpose of determining the quantity of steam consumed by each apparatus, we carried out special trials on the spot.

We thus arrived at the following consumption figures :

Atomiser.	Steam presssure at which the devices functioned.		Consumption of saturated steam per hour.		Consumption as a percentage of the total steam production.
	kgr./cm²	*lb./sq. in.*	kgr.	*lb.*	
Serbescu-Grecu	4.3	*61.1*	250	*551*	4 %
Cosmovici	3.5	*49.8*	320	*705*	5 %
Marsi	5.0	*71.1*	450	*992*	7 %

We used the results of the foregoing trials to estimate the consumption of saturated steam required for the auxiliary services, and this gave us the following :

2 % for warming the mazout;
2 % for driving the air brake pump;
3 % for warming the train;

8 % exhaust steam taken for working the exhaust steam feed-water injector.

As we used two kinds of fuel, viz. : mazout and lignite, and in order to ascertain the results obtained, we converted these fuels into a fuel-standard called « *Cardiff coal* », which would have a heating value of 7 500 calories. When compared with this fuel-standard, we found that 1 kgr. of mazout is equivalent to 1.3 kgr. of Cardiff coal, and 1 kgr. of lignite to 0.5 kgr. of Cardiff coal.

We also made allowance for the fact that the steam was superheated differently by the various devices, by converting the superheated steam into normal steam containing 639 calories (2 560 B. T. U.).

The trials were carried out on two types of locomotive, viz. : The *Pacific* locomotive (231.000) and the P.8 locomotive (230.000).

The results are set out in the tables 1 and 2 below.

After trying the atomisers on the *Pacific* locomotive, we found that only the *Marsi, Popescu* and *Serbescu* atomisers shewed a saving as compared with the *Cosmovici* atomiser. The *Molnar* atomi-

Table 1.

Trials of « Pacific » locomotive No. 231.070 between Bucharest and Pitesti (108 km. 67.1 miles) with a fast train.

The train was heated; the boiler had a heating surface of 250 m² 2 690 sq. ft.), a pressure of 15 atm. 185 lb. per sq. in.), and was fed by a saturated steam injector of the « Friedmann » type.

Type of atomiser.	Date of trial.	Weight of train, in tonnes.	Journey time, in hours.	Consumption of fuel, in kgr.			Consumption of water, in m³.	Temperature		Boiler efficiency, %.	Boiler output, in kgr./m² per hour.	Weight of steam obtained from 1 kgr. of Cardiff coal.	Average values.
				Mazout.	Lignite.	Cardiff coal.		Water, °C.	Superheat, °C.				
Molnar	7. XI. 1935	512	2.07	1 177	1 260	2 260	16.770	15	265	63.8	32.6	7.47	7.81
»	9. XI. 1935	511	2.07	1 163	1 080	2 135	17.270	16	265	69.5	33.7	8.15	
Cosmovici	12. XI. 1935	513	2.07	1 172	1 224	2 230	16.880	15	265	65.0	33.0	7.65	7.40
»	13. XI. 1935	514	2.07	1 172	1 116	2 165	15.450	13	265	61.3	30.0	7.15	
Gh. Popescu-Lemaitre	16. XI. 1935	514	2.00	1 035	1 116	1 985	15.660	14	265	67.8	31.6	7.95	7.97
»	18. XI. 1935	510	2.07	950	1 116	1 875	14.900	14	265	68.3	29.0	8.00	
Marsi	21. XI. 1935	514	2.07	1 040	1 116	2 005	15.510	11	265	66.0	29.8	7.72	7.73
»	22. XI. 1936	520	2.07	1 040	1 188	2 040	15.890	12	265	66.0	30.5	7.73	
Serbescu-Greeu	26. XI. 1935	520	2.09	884	1 170	1 830	15.260	11	265	73.0	28.8	8.55	8.47
»	28. XI. 1935	514	2.07	925	1 170	1 880	15.430	11	265	71.5	30.6	8.40	
Jelinek	4. XII. 1935	520	2.08	1 095	1 188	2 110	15.560	13	265	64.0	27.7	7.50	7.42
»	5. XII. 1935	518	2.07	1 172	1 224	2 230	16.260	13	265	62.8	31.7	7.35	

TABLE 2.

Trials on P. 8 locomotive No. 230.062 between Bucharest and Pitesti (108 km. = 67.1 miles) with a fast train.

The train was heated; the boiler had a heating surface of 144 m² (1 550 sq. ft.), a pressure of 12 atm. (170.7 lb. per sq. in.), and was fed with water by an exhaust steam injector of the « Friedmann » type.

Type of atomiser.	Date of trial.	Weight of train, in tonnes.	Journey time, in hours.	Consumption of fuel, in kgr.			Consumption of water, in m³.	Temperature		Boiler efficiency, %.	Boiler output, in kgr./m² per hour.	Weight of steam obtained from 1 kgr. of Cardiff coal.	Average values.
				Ma zout.	Lignite.	Cardiff coal.		Water, °C.	Superheat, °C.				
Cosmovici (atomiser on locom. 230.262) (*)	27. III. 1936	546	1.93	910	900	1 700	12.040	15	300	63.0	45.5	7.38	
Cosmovici (atomiser on locom. 230.262)	28. III. 1936	528	1.95	900	041	1 770	13.125	13	300	66.2	47.8	7.76	
Cosmovici (Steam nozzle, 0.7 mm.)	30. III. 1936	524	1.94	790	1 076	1 650	12.350	14	305	66.7	47.6	7.81	7.60
" "	31. III. 1926	525	1.95	1 000	694	1 700	12.200	12	300	64.0	45.6	7.50	
Cosmovici (Steam nozzle, 1 mm.	2. IV. 1936	529	1.95	960	867	1 760	12.650	12	295	64 0	47.1	7.50	
	3. IV. 1936	527	1.94	940	970	1 780	12 850	14	300	64.5	48.2	7.65	
Serbescu-Greco (*)	21. IV. 1936	525	1.91	760	868	1 500	12.012	14	290	72.5	46.3	8.50	
" "	22. IV. 1936	527	1.92	1 000	729	1 720	13.075	13	305	68.7	50.2	8.05	8.14
" "	24. IV. 1936	514	1.94	960	1 110	1 890	14.120	10	310	67.8	54.0	7.95	
" "	25. IV. 1936	518	1.93	780	1 145	1 670	12.640	14	300	68.7	48.4	8.05	
Marsi	27. IV. 1936	521	1.95	860	1 110	1 660	12.455	21	300	64.7	44.7	7.58	7.44
"	28. IV. 1936	519	1.95	1 080	954	1 950	13.950	19	310	62.2	50.8	7.30	

Note. — (*) Train not heated.

ser proved to have a greater evaporating capacity than the *Cosmovici*, but the saving resulting from this was not as large, in view of the heavy steam consumption necessary to effect the atomisation.

The *Popescu* atomiser was withdrawn by the maker after the trials had been carried out on the *Pacific* locomotive; and thus there only remained the *Cosmovici*, *Marsi* and *Serbescu* atomisers for the trials on the P.8 locomotive.

In order to obtain supplementary information about the qualities of the *Cosmovici* atomiser, we tested 3 kinds of atomiser of this make, each having different slits for the passage of the steam, ranging in size between 0.7 and 1 mm. (0.027 and 0.0394 in.).

Comparative results.

The boiler hourly output ranged between 44.7 and 50.8 kgr. per m² (9.15 and 10.36 lb. per sq. ft.) of heating surface for the P.8 locomotive, and between 27.7 and 33.7 kgr./m² (5.67 and 6.90 lb. per sq. ft.) for the *Pacific* locomotive. On one of the trials, the boiler output was 54 kgr./m² (11.06 lb. per sq. ft.), the brakes on one of the coaches in the train having become applied en route, due to an irregularity in working.

In Table 3, we have given the comparative results obtained from the point of view of evaporative capacity and economy, in relation to the *Cosmovici* atomiser.

TABLE 3.

Classification in order of evaporative capacity.

No.	Atomiser.	Evaporation in kgr. of normal steam per kgr. of Cardiff coal. Locomotive			Percentage of economy in relation to the Cosmovici atomiser.
		231.070	230.262	Average.	
1	*Cosmivici.* . . .	7.40	7.60	7.50	—
2	*Jelinek.*	7.42	—	7.42	— 1
3	*Marsi*	7.73	7.44	7.58	+ 1
4	*Molnar.*	7 81	—	7.81	+ 4.13
5	*Popescu*	7.97	—	7.97	+ 6.27
6	*Serbescu*	8.47	8.14	8.30	+10.60

It will be seen that the *Serbescu* atomiser gave the greatest economy, which is about 10 %.

The *Serbescu* atomiser gave a better performance on the *Pacific* locomotive, which has a wider firebox than the P.8 locomotive, whilst the *Cosmovici* atomiser is more suitable for the narrow firebox.

[625. 245]

Improvements in the construction of tank wagons,

by Hans RICHTER, Chief Engineer, Hamburg.

The tank wagon for the transport of liquids has now become an indispensable means of conveyance for a number of purposes. All manner of liquids, such as acids, lye, carbon disulphide, etherised oil, paraffin, petrol, benzol, toluol, gas oil and other liquid fuels, dietetic oils, wine, alcohol, milk and many other products are carried in tank wagons from the producer to the consumer, or from the point of extraction to the factory for transformation.

Continually increasing motorisation has particularly contributed to make the tank wagon one of the most important means of transport. Besides railway tank wagons there are nowadays road motor tanks, fitted with all improvements, which serve, with the greatest speed, the extensive area round the distribution points.

* * *

It is needless to say that the great importance assumed by the tank wagon has caused to increase more and more the conditions it is required to fulfil, particularly as on account of the ever-increasing quantities of very highly inflammable and explosive liquids, it is also necessary to take adequate measures for a high degree of security for the public and staff. It has, naturally, been especially necessary to impose rigorous regulations with regard to volatile liquids.

The new types of tank which have been designed for this particular kind of service are certainly of great importance for all other traffic requiring the use of tank wagons.

Many other liquids carried by tank wagons, such as acids, lye and other chemical products have dangerous properties which necessitate the adoption of all means available for guaranteeing the safe utilisation of this stock.

A typical railway tank-wagon of the old type of construction is shown in figure 1. The superstructure of this wagon has remained the same, essentially, for some decades, apart from various special devices which have had to be adopted at different times owing to special regulations of the Reichsbahn.

In order to judge the safety in service of a liquid-storage wagon, it is necessary to consider, above all, the drainage arrangements, the frequent use of which imposes a large amount of wear and tear and also causes a tendency to develop leaks : this is the more important as they are fitted at the lowest point of the reservoir, where leaks give rise to the greatest dangers.

The quality of this accessory therefore serves as a criterion for that of the whole tank — apart from the properties of the frame and running gear, which form a chapter apart. This fact having been recognised, the evolution of tank wagons has now commenced by the remodelling of the drainage apparatus, after many years of inaction in this domain.

In the tank wagon shewn in figure 1 the drainage device consists simply of a plug bedded in the bottom of the tank and pressed on its seat by a long bar from the top of the cylinder. It is flanked on both sides by discharge tubes with taps at the ends by which, if the plug is

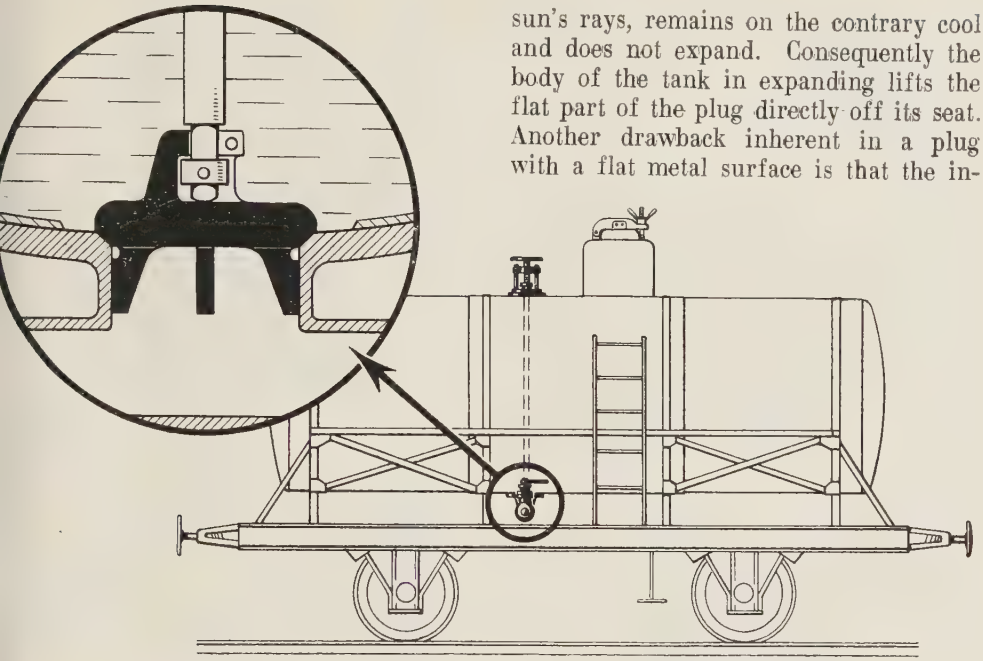

Fig. 1. — Old type railway tank wagon, having single-cone bottom valve with metal joint (shewn top-left on a larger scale) and lateral discharge taps.

lifted, the contents of the tank may be extracted. These arrangements, dating from a time when tank wagons played only a secondary part, are not sufficient to meet present requirements. It is known now that this system, in which the metal valve plate is pressed against its seat by a long bar from the top of the tank with a closing screw placed on the tank, or in the dome, arises from an entirely erroneous conception. It is necessary to realise, in fact, that a tank wagon is exposed to great variations in temperature, when, for example, it is taken from the shade into strong sunlight, which it is impossible to avoid. In this case the outside casing of the tank is heated and expands considerably. The plug-operating bar, which is inside the tank and not subject to the action of the sun's rays, remains on the contrary cool and does not expand. Consequently the body of the tank in expanding lifts the flat part of the plug directly off its seat. Another drawback inherent in a plug with a flat metal surface is that the introduction of small foreign bodies (hammer-scale, for example) between the surfaces of the joint is sufficient to render it inoperative on the whole of its circumference.

Leaks occurring between the two faces of a metal joint as a result of the presence of a small foreign body are of great importance. For example, if a particle only 1/4 mm. (0.01 in.) thick sticks on the face of a plug of 100 mm. (0.39 in.) interior diameter, the cone of the closing plug is pressed obliquely against its seat. The resultant leak forms an angle whose transverse section is 36.17 mm² (0.0039 sq. in.), and in one hour 350 litres (77 Br. gallons) of petrol can be lost through this leak, with two metres (6 ft. 6 3/4 in.) depth of liquid in the tank, which is a very considerable quantity.

The taps at the ends of the discharge tubes have also caused much trouble. A tap is, even under normal conditions, a very uncertain means of closing and should not be employed at all for volatile liquid fuels. These fuels, such as petrol, benzol, etc. dissolve all lubricating greases, so that the key of the tap rubs dry in the socket. When the tap is used grooves are formed in the surfaces of the conical joints, so that even at the end of a couple of weeks considerable leaks can occur, and the key must be continually reground. Also the jamming of the cock entails for the staff the use of a hammer to open the tap, which is another drawback.

* * *

Figure 2 shows a modern tank wagon provided with the equipment required by present regulations. It has a pressure compensator and modern safety sieves, which are more effective than the simple Davy metallic gauze against back-firing, electrical earthing for protection against electrostatic charges, and a wave-breaker for protection against the to and fro movements of the contents. Progress is evidenced above all, however, in the improvements in the drainage apparatus. Instead of the simple bottom valve with single plate, there is a two-faced valve with independent joints moving in contrary directions, which, instead of being forced on their seats from the top of the tank, are operated by a threaded stem which joins the two faces. This is already a great improvement on the old simple bottom valve. The stem presses the two plates like two jaws of a vice against the body of the double surfaced valve-seat separating them, so that the double closing required by the regulations is really obtained.

It is opportune again to call special attention to the fact that the two plates close quite independently of one another; in other words, if by some rare chance

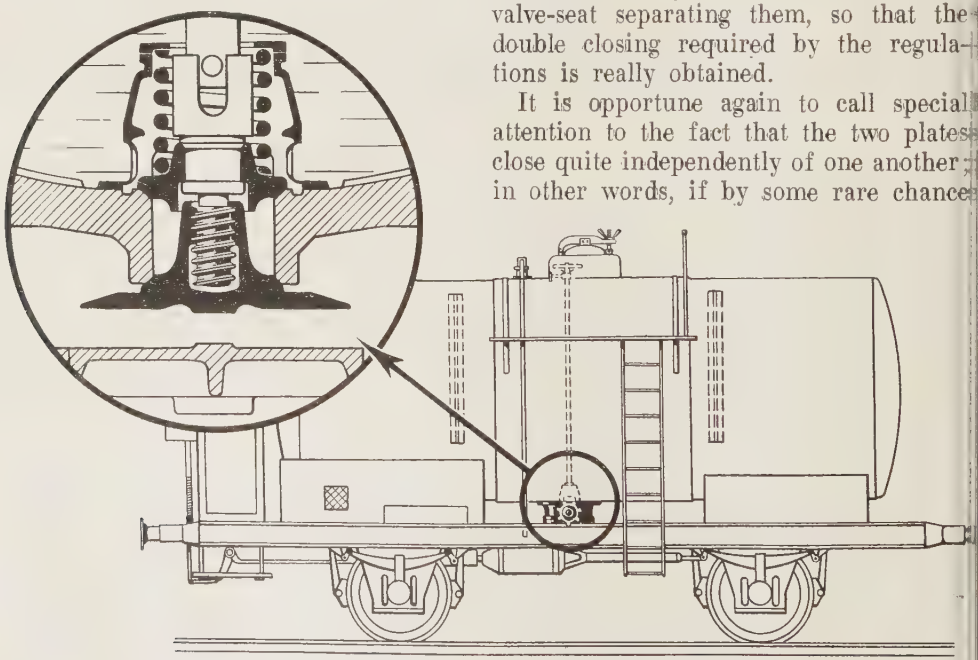

Fig. 2. — Modern railway tank wagon with double-cone bottom valve and metal joints, so that closing caps are sufficient for the discharge pipes.

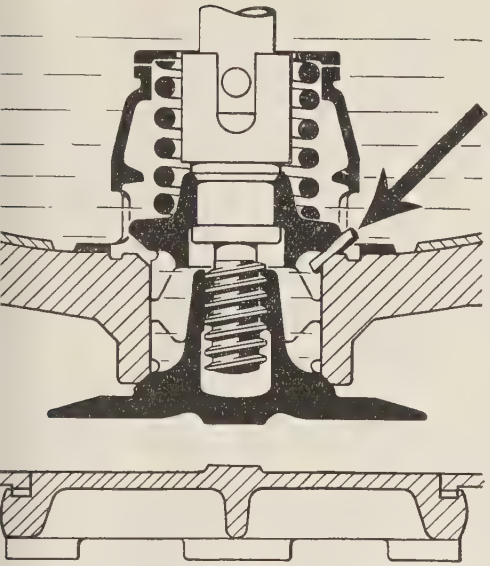

Fig. 2*a*. — Both plates of the bottom valve in figure 2 close quite independently of each other, so that a tight fit is assured even when one of the plates, for any accidental reason whatsoever, does not operate.

one of them is prevented from closing (for example, by the insertion of a large foreign body) the other is nevertheless closed in perfect fashion, so that a sure closing is guaranteed.

This double independent closing is of prime importance, because it ensures that, even in the case of large impurities or introduction of foreign matter, there is still a sure closing. In order to realise this independent closing of the two plates, they are mounted freely moveable along the axis of the shaft which joins them. They are therefore free to move until they meet an abutment in exactly the same way as the two jaws of a vice are capable of imprisoning a small object as well as a large one, without affecting the force of closing. At the same time the unsatisfactory transmission of the effect of closing from the top of the tank has been suppressed, so that the stem of the valve can no longer become bent, which moreover, would have no influence on the tight closing of the bottom valve, since the threaded shaft is entirely independent of the valve stem. This latter item is of such massive dimensions and robust construction that any deterioration is quite impossible. The tightness of the joint cannot be compromised, moreover, by vibrations in traffic or any deformations of the tank barrel. The double closing is so efficient that the heavy taps are no longer necessary at the ends of the discharge pipes, but it is sufficient to fit them with caps to protect their threads.

As it is absolutely necessary to be sure there is no dripping of the inflammable, corrosive or toxic contents of the tank, with the bottom valve described above, the plates used have to be so designed that the introduction of small foreign bodies, such as hammer-scale, particles of rust, etc., between the surfaces of the joint cannot prevent its tight closing. As this condition cannot be fulfilled by metallic obturating surfaces, recourse has been had to the elastic joint shewn in figure 3.

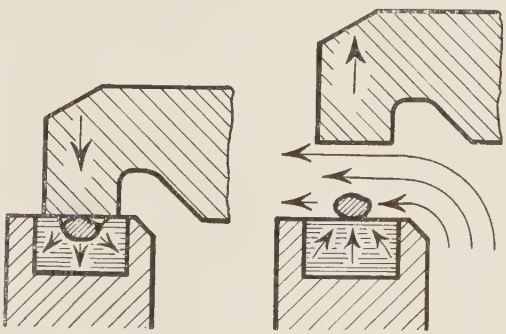

Fig. 3. — Method of operation of elastic joint in modern bottom valve.

The practical application of this system, however, is subject to various difficulties, because no material exists which is at the same time elastic like rubber and impervious to the action of petrol, benzol, oil and acids. Rubber itself is dissolved in volatile oil. Other materials by which is was hoped to replace rubber possess equally unsatisfactory properties; they are, for example, soluble in water and so are attacked by the water contained in the liquid fuel. Or they may have the drawback of drying up in time and becoming hard and brittle.

It was therefore necessary to find an entirely new jointing material, combining a whole series of different properties normally incompatible with each other.

The discovery of this material can be considered as marking a step in the evolution of those drainage devices, since from the moment the former metallic joint could be replaced by an elastic joint, a certain number of important advantages were obtained. Whilst the metallic joint is notably affected by the penetration of foreign matter between the surfaces, the new elastic joint is practically insensible to their presence, since they simple bury themselves in the elastic material of the joint and are automatically thrown out when the valve opens, and swept away by the flow of liquid (fig. 3). Another advantage of the new elastic « Durogum », is that the joint for which it is used always closes with a really perfect tightness, because each of the deteriorations to which a ground metal joint is exposed such as the formation of grooves, hollows and scratches cannot occur with this material.

With metal joint surfaces it may happen that even a few hours after the mounting of the drainage device a grain of sand produces a serious leak which necessitates the emptying of the tank wagon and the re-grinding of the valve face.

Previously, in spite of improvements effected, there was still the considerable drawback that in order to re-establish the tightness of the bottom valve it was necessary for someone to enter the tank. Whilst according to the conditions laid down to prevent accidents the tank must be carefully ventilated, this descent into the tank always presented grave risks for the staff. Also, the repairs naturally could only be made inside the tank in a summary manner, since his cramped position restricted the workman's movements. Figure 4 shews the very simple way in which these difficulties have been overcome. The shutting-off device is simply mounted in the case rivetted to the tank case so that it can be dismantled from below, that is to say without entering the tank (figure 4). With this construction the lower plate of the valve can be dismantled even from a full tank (fig. 5) because during this time the large plate is closed by a strong spring. Consequently, if a leak appears after the tank wagon is filled, it is no longer necessary to have it pumped out, but simply to dismantle the smaller plate of the valve and fit new stopping rings, so that the bottom valve again provides a tight closure, until it is possible to dismantle also the larger disc, serving as a safety closing.

* * *

Needless to say, the progress effected with regard to rail tanks has also been applied to road-motor tanks. However by reason of the special conditions which vehicles of this type must fulfil for road transport, several supplementary con

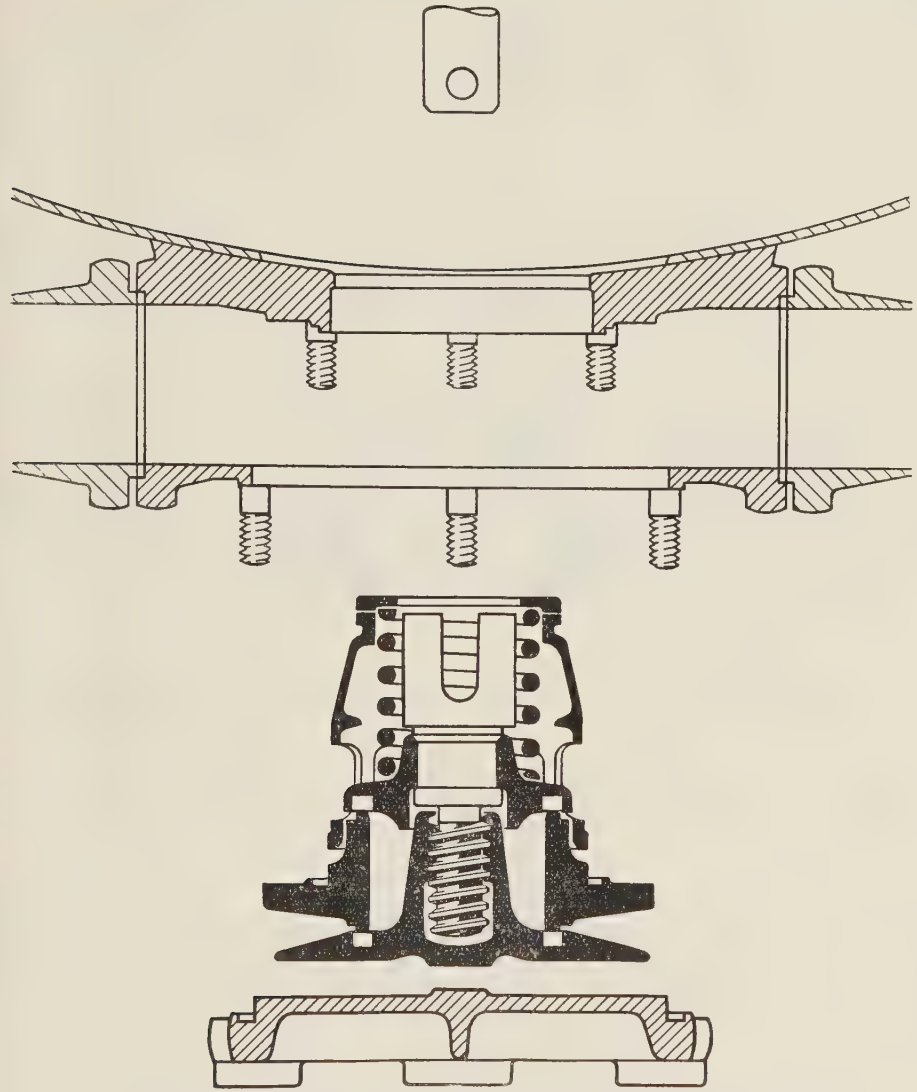

Fig. 4. — Modern bottom valve, double-cone, with cut-off apparatus, removable from below, i.e. without any necessity for a man to enter the tank.

struction details have been necessary. Faults in the drainage system can give rise to particularly severe accidents in the case of vehicles working on the roads.

There are also more serious risks. For example, the threaded joint by which the drainage hose is fixed to the outlet valve may easily leak. Then, if by ill-chance, some careless passer-by should throw down a match and set alight the combus-

tible liquid which has dripped out, it would rarely be possible to close the tank outlet-valve, situated in the immediate vicinity of the threaded joint and exposed to the flames. Further, it would be necessary to act quickly, and the screwing down of the bottom valve, if still possible, would take too long.

It is, then, very advisable to provide road-motor tank vehicles, as shewn in figure 6, with a supplementary rapid-closing device, operated by a cord from the driving cab or other suitable part of the vehicle, for the instantaneous and rapid closing of the bottom valve (fig. 7). A system of this type, already legally prescribed in Holland and Denmark, may include the use of a fusible metal wire or other similar means of automatically ensuring immediate closing in case of fire.

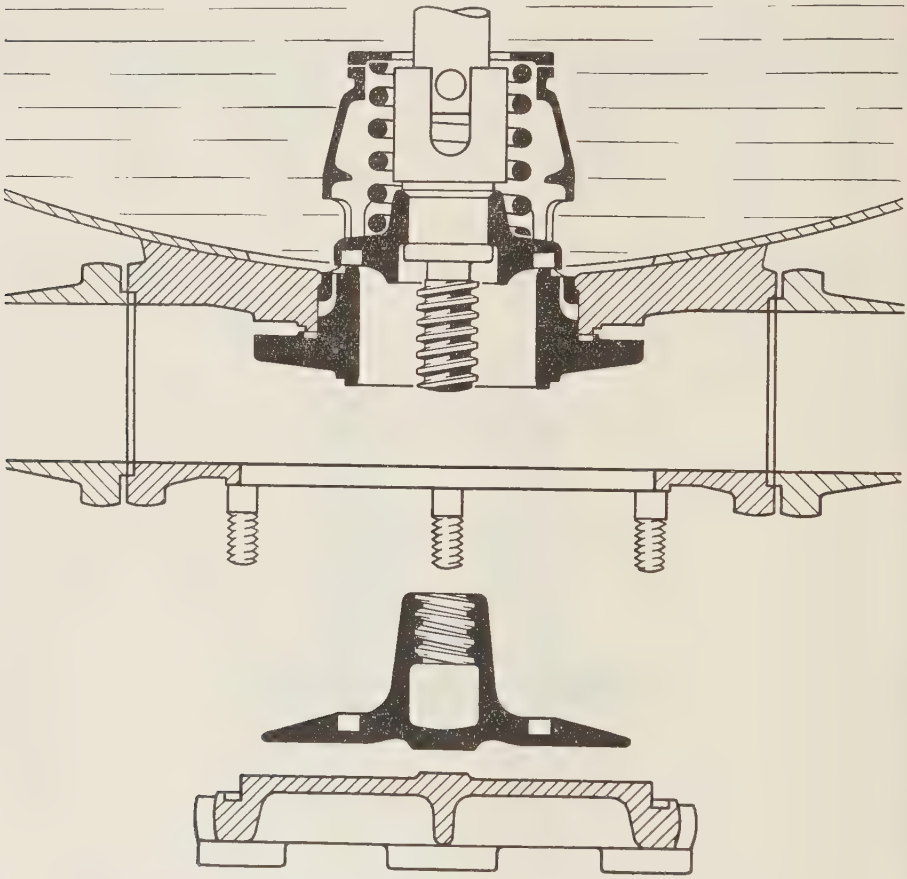

Fig. 5. —With the modern bottom valve with double cone, the lower cone can be dismantled from a full tank, and fitted with a new packing ring.

A modern road tank, thanks to its

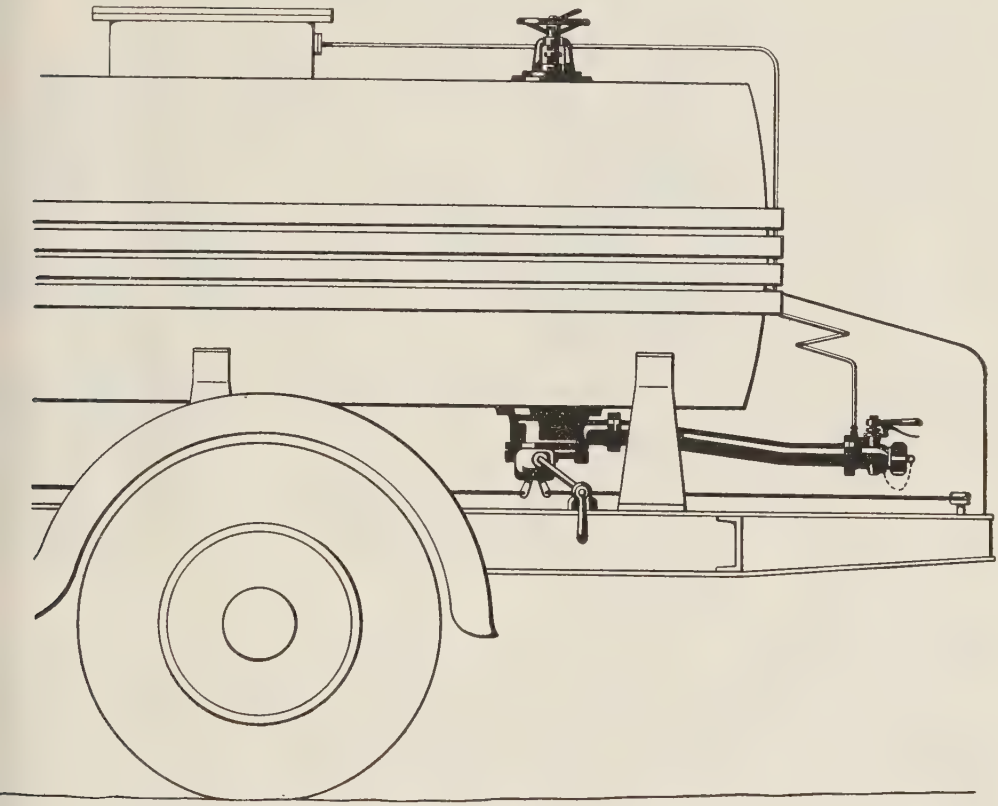

Fig. 6. — Road motor tank with bottom valve equipped with a device allowing of immediate closing in case of fire, etc.

equipment, thus offers the very large guarantee of security absolutely necessary for road traffic on account of the large quantities of inflammable material transported. But the modernisation of tanks is not confined to bottom drain cocks; the other devices such as outlet taps, distributors, etc., have undergone a remarkable evolution which has made the tank vehicle a sure, powerful and ever-serviceable vehicle.

We have already pointed out that discharge cocks are a very unsatisfactory means of obturation. On the other hand, in many cases it is necessary to have some cutting-off device at the ends of the discharge pipes, — for example, for drawing off small quantities. In order to provide a safe device for this need also, the principle of the elastic joint has again been applied to the discharge apparatus, in the manner which proved so satisfactory with the bottom valves.

Figure 8 shows that it has been possible to design a special device which combines the theoretical advantages of a tap with the practical advantages of a valve. A tap, theoretically, has the advantage of quick and easy operation by means of a simple lever (in practice this advantage

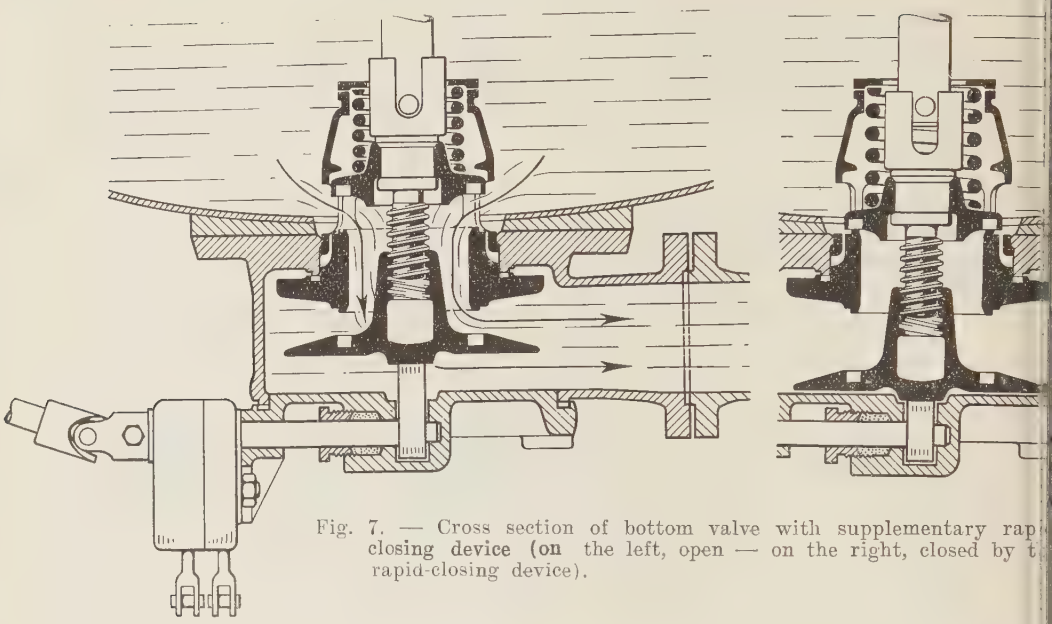

Fig. 7. — Cross section of bottom valve with supplementary rapi closing device (on the left, open — on the right, closed by t rapid-closing device).

does not exist, the key of the tap being generally too difficult to turn). However, the ease of operation which is sought with the tap does exist, not only theoretically but actually, with the drain cock shown in figure 8. A cutting-off cone, arranged horizontally, is operated by an eccentric, connected to an operating lever. The large conical surfaces of the cock, the grinding of which required much time and a very skilled and careful workman, are replaced in this type of valve by elastic stopping rings, which are easily renewable.

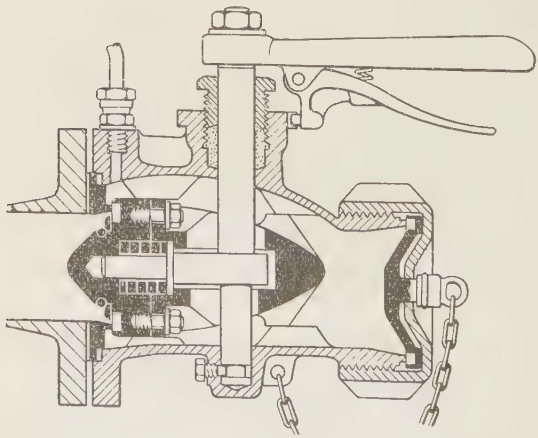

Fig. 8. — Modern rapid-extraction tap, operated by a lever with cut-off cone, arranged horizontally, and elastic joint.

This cock also is an appreciable improvement on the so-called easily-worked taps. In practice, the easy operation of these taps is obtained by giving play to the key, thus creating an artificial gap, into which minute foreign bodies can penetrate, so tending to create grooves. In addition, the essential drawback of large frictional obturating surfaces is also present in these taps.

The governing principle in the drain cock shewn in figure 8 is suitable for various other applications. It is first of all possible, on the basis of this principle, to realise multi-way distribution,

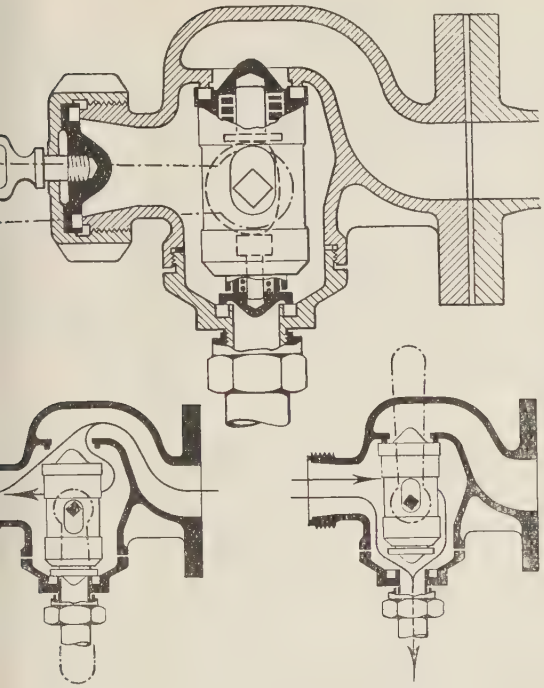

Fig. 9. — Two-way extraction tap, with coupling for drainage hose pipe.

allowing by a single movement of the lever the establishment of different directions of flow, for which a series of valves operated alternately in a determined order of succession had previously been necessary. Figure 9 shows the excellent way in which, for example, two-way distribution can be obtained, allowing a tank at a high level to be filled, and then by simply advancing the lever to stop the flow and at the same time to empty the liquid remaining in the hose pipe.

Figure 10 shews a very interesting solution of a characteristic multiple-distribution problem. The body of the valve

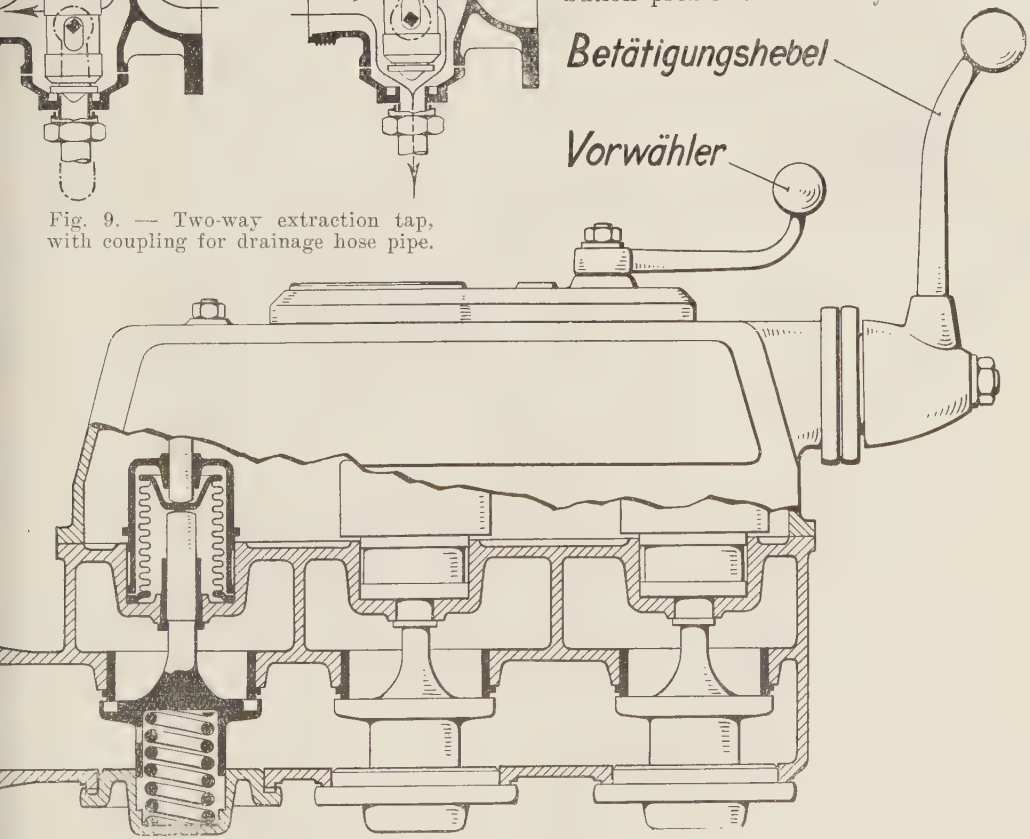

Fig. 10. — Multiple distribution, with selector.

Note. — Betätigungshebel = operating lever. — Vorwähler = selector.

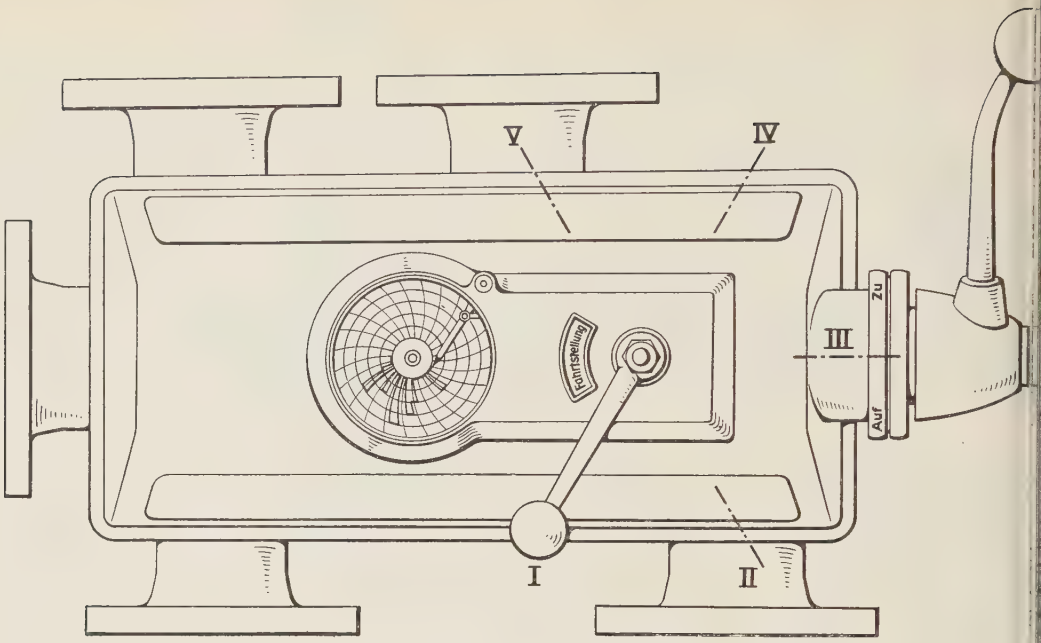

Fig. 11. — Plan of multiple distributor with recording apparatus in position.

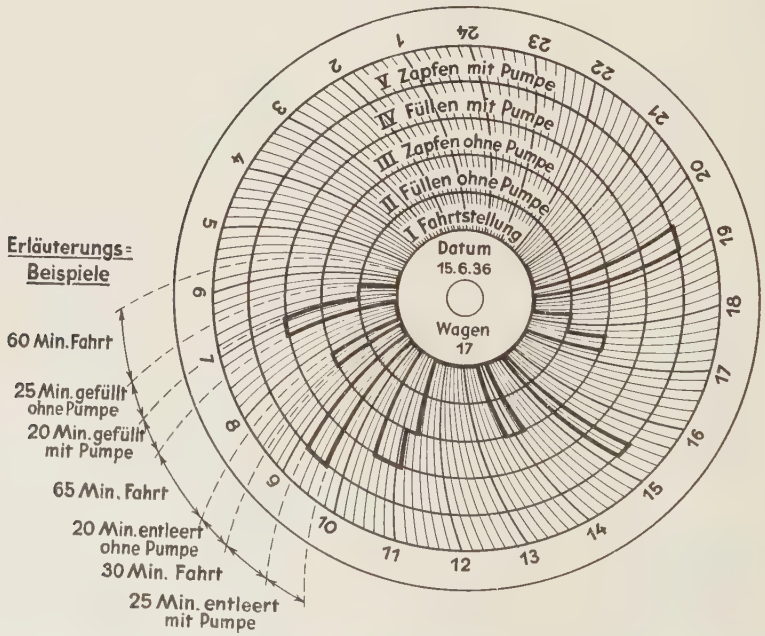

Fig. 12. — Diagram sheet from recording apparatus of multiple distributor.

Translation of German terms (top to bottom). — 60 minutes travel. — 25 minutes filling without pump. — 20 minutes filling with pump. — 65 minutes travel. — 20 minutes drainage without pump. — 30 minutes travel. — 25 minutes draining with pump.

V. Extraction with pump. — IV. Filling with pump. — III. Extraction without pump. — II. Filling without pump. — I. Running position. Date. Vehicle.

is provided with a selector by means of which, for example, that section of the tank (in a tank wagon with three reservoirs) which is to be emptied can be determined. In addition, the direction of the flow can be established as desired, that is to say, the manner in which the draining-off shall take place — at the side or at the rear; the apparatus can also be set in the filling position and in this case the selector can also be used to determine which reservoir shall be filled.

In order that the head man, for statistical purposes, can find out exactly when and how long the distributor has been in action, and which distribution has been used, it is possible to provide the multiple distributor with a recorder (fig. 11), by which each setting in action of the distributor is recorded on a diagram sheet (fig. 12).

* * *

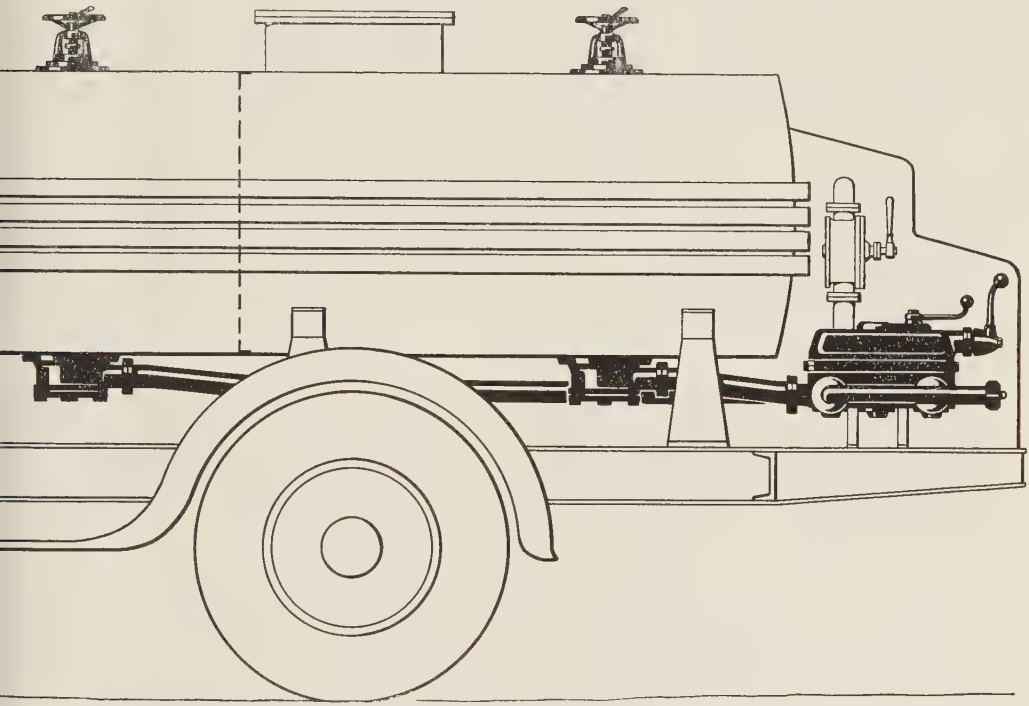

Fig. 13. — Road motor tank with modern bottom valve and multiple distributor, replacing all the individual valves formerly necessary.

The foregoing notes give an idea of the degree of improvement effected in the tank vehicles, and particularly in the means of drainage. In this way it has been possible to overcome many former difficulties, and above all to realise also a very considerable progress, from a technical point of view, in the protection against accidents.

It is hoped that these efforts to impro-

ve tank vehicles unfortunately only recently taken in hand, will be continued assiduously, in order that the successful progress of motorisation may not be trammelled by defects in the technical devices necessary for the transport of liquid fuels. It is essential that the consumption of liquid fuels and the possibilities of transporting them should be so harmonised that the conveyance and service to consumers of liquid fuels can be effected without difficulty.

The progress realised in this domain will therefore be of value also to other industries which use tank vehicles.

[656. 212]

Improvement in the sorting of post parcels at the Brussels-Midi station concentration office,

by A. NIZET,
of the Belgian National Railways Company.

General organisation of the post parcels service.

The post parcels service in Belgium s been carried out for the past five ars by the Belgian National Railways ompany, which is responsible for the ıole of the organisation as well as con- yance, the Post Office collaborating for e acceptance of parcels and also for livery at places not reached by rail or rtage services.

Parcels not despatched direct to des- ıation for same-day delivery are dealt th in two phases, the first, the end- the day phase, consisting in their con- ntration at 33 offices where they are rted, packed into bags and forwarded destination. The second phase com- ences first thing the next morning ıen the concentration points are trans- rmed into distributing offices; they en distribute the parcels in their areas th a view to delivery by 1.0 p.m. the me day.

Owing to its situation, the Brussels- di station houses by far the largest rting office in the country, dealing th more than half of the parcels car- ed. Equipment designed for a capacity 5 000 to 6 000 parcels was installed a w years ago and proved satisfactory un- recently. It consisted principally of a ng zinc-covered table on which the par- s were discharged and sorted, sur- unded on three sides by wooden racks in which they were placed in readiness for packing into bags. Figures 1 and 2, and the sketch, figure 4, give an idea of the lay-out.

The installation and its personnel were subsequently enlarged to meet the steady growth in the number of parcels, but eventually the daily volume rose to more than double its former level with consequent congestion and risk of damage and errors in sorting.

This, together with a desire to improve the general efficiency of the service by reducing as much as possible the number of concentration points, led the Company to seek a solution in the partial mechanisation of the sorting operations. It was therefore decided, in view of the favourable situation of Brussels-Midi station, to extend the activities of the sorting office at this point, and at the same time to reduce the number and size of the concentration offices in the South-West of the country (see sketch, fig. 3).

New organisation.

In considering what mechanical arrangements to adopt for the handling of parcels the Company had regard to what had already been achieved in this direction in other parts of Europe, whilst endeavouring to reduce to a minimum the defects noticed in the operation of the installations in service at the time. There was also the desirability of adapting the in-

Fig. 1. — Former sorting installation for post parcels.

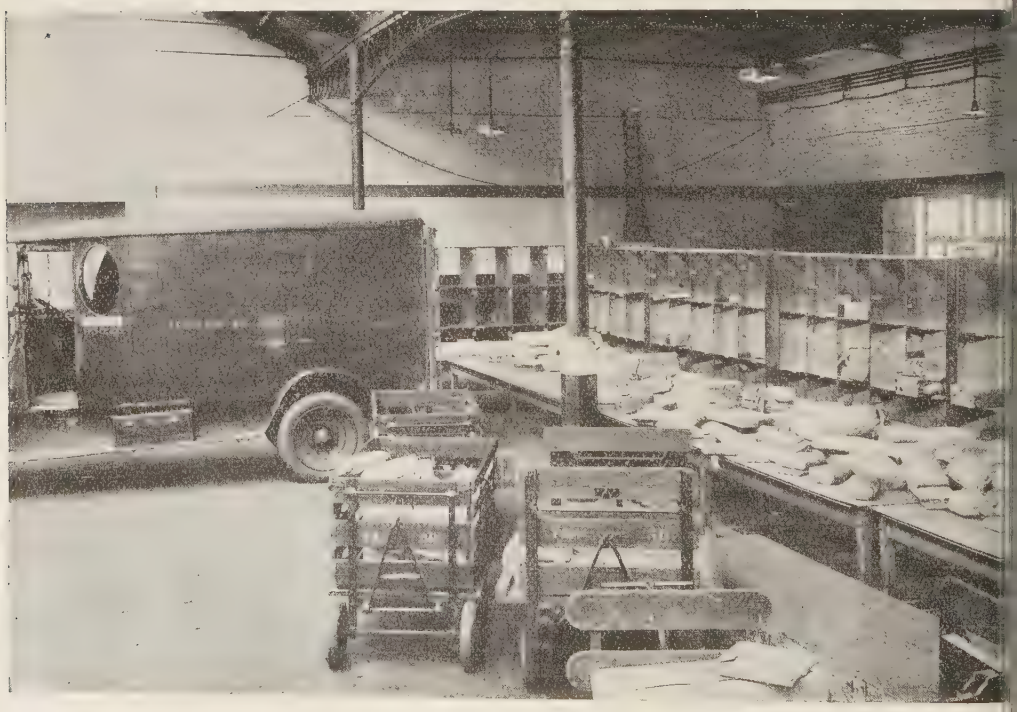

Fig. 2. — Unloading vans and tricycles.

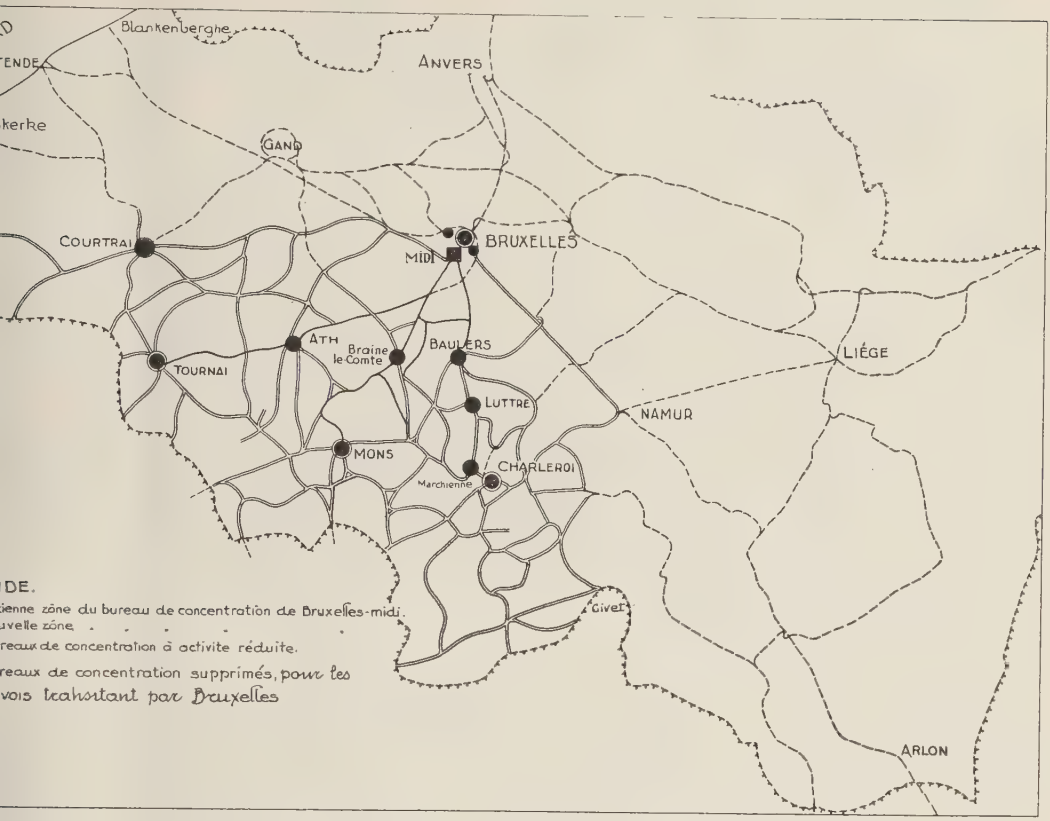

Fig. 3. — Zone covered by the Brussels-Midi concentration office.

Explanation :

Former zone of Brussels-Midi sorting office.
New zone of Brussels-Midi sorting office.
Sorting offices whose activities are reduced.
Sorting offices abolished for parcels transiting via Brussels.

allation in part to the existing general rganisation.

One feature of the Belgian post parls organisation which particularly lent self to mechanical sorting was the arking of parcels in bold characters on ceptance with a letter and number code indicate the destination office, this ing in order to facilitate the several ases of sorting necessitated by the parls service.

The lay-out shown in figure 5 was eventually designed and installed. The 7 000 to 8 000 parcels collected from the accepting offices in and around Brussels, already marked with their code numbers, are brought in by motor van to the Brussels-Midi sorting office and discharged, as they arrive, at C1, C2 and C3, on to the conveyor belt I_1, travelling thence via belts I_2, I_3 and I_4 to the distributor ring which, by continuously re-

volving, carries the parcels round in single file in front of the sorters standing inside on stage D, who merely have to push them over on to one or other of the 12 belts E, each of which covers a group of destinations.

Any parcel that escapes the vigilance of a sorter automatically travels round until it reaches him again.

The parcels proceed on the belt and roller conveyors E towards the 186 racks F, in which they are placed by sorters as they reach the tables numbered 1 to 12. The ordinary parcels are afterwards packed into the bags G by hand, whilst those of a fragile or bulky nature are placed uncovered on tricycles or into metal basket containers.

Bags, containers and tricycles are finally evacuated to trains and motor vans for conveyance to destination.

Parcels accepted locally at Brussels-Midi, and also those brought in by train, are taken by tricycle to B where they are placed on the belt conveyor I_2.

Four to five thousand parcels collected at domicile in Brussels and its environs, which do not pass through a local accepting office, have to be marked with the code designation representing their destination office. They are therefore discharged at A next to the markers, by whom they are subsequently deposited on the conveyor I_2. All the parcels placed on this conveyor follow the same route via I_3, I_4, E, etc., as that described above in connection with those discharged at C on to conveyor I_1.

The equipment was installed by a Belgian firm specialising in the design of mechanical handling plants (see figs. 6, 7 and 8).

All the conveyors are 700 mm. (27 1/2 in.) wide and all their working parts are mounted on ball or roller bearings. The roller conveyors consist of a double train of rollers 40 mm. (1 9/16 in.) in diameter spaced at 5-mm. (2 in.) intervals.

The pairs of rollers slope in towards each other so as to check the tendency of cylindrically-shaped parcels to roll off. The same arrangement is adopted for the belt conveyors, the cotton fabric belts being curved inwards by means of double sets of large-diameter rollers.

The distributor ring consists of an annular steel plate having an outside diameter of 10 m. (32 ft. 9 3/4 in.), supported by 16 cast-iron rollers. Motion is imparted by a 1.5-H.P. electric motor through a worm reduction gear and a rubber-tyred friction wheel which contacts the plate from below, good adhesion being obtained by mounting the motor and reduction gear on a rocking frame fitted with a counter weight.

The belt conveyors are each driven by a separate motor with worm reduction gearing, and although their total length is 175 m. (574 ft.) the current consumption is not large, the total power of the 12 motors being only 7.5 H.P.

To facilitate operation of the various conveyors distant-control equipment is installed, the motors being started and stopped from push-buttons concentrated inside the distributor ring within reach of the team of sorters.

The fitments surrounding the installation were the subject of very careful consideration, and are designed to allow maximum speed with a minimum risk of damage.

The three-stage racks have a total length of 62 m. (203 ft. 5 in.), giving 186 compartments of 450 dm^3 (16 cu. ft.) capacity each, and are constructed of strong standardised elements made of

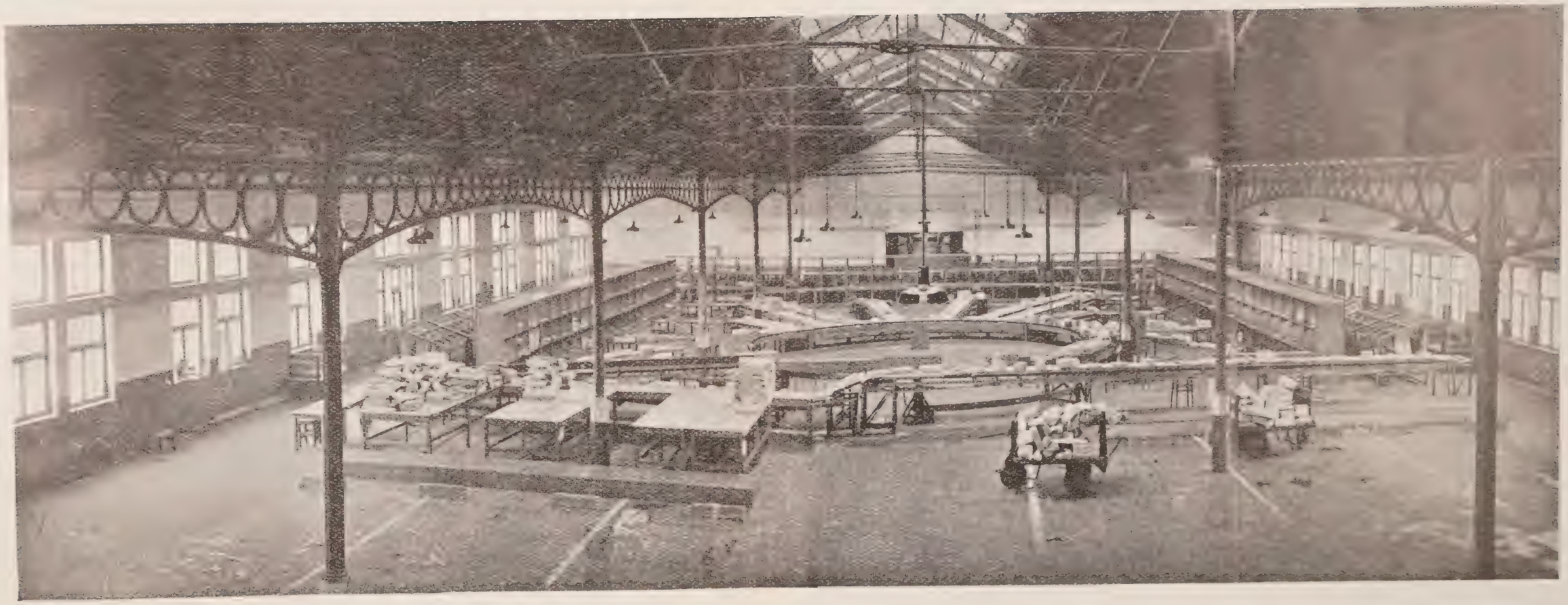

Fig. 6. — General view of the new sorting office.

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Fig. 7. — (1) Panoramic view of the installation.
The parcels are discharged on to the right-hand conveyors and taken off the left-hand conveyors in front of the racks.

Fig. 8. — (2) Panoramic view of the installation.
Seen from this side, the parcels are moving from left to right. In the foreground is the enclosed automatic switch gear of the installation. The 12 motors are started and stopped by distant control from the push-buttons visible in figure 7, inside the distributor ring, on the left.

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steel, which permit of several different combinations.

The bags are hooked in groups of 12

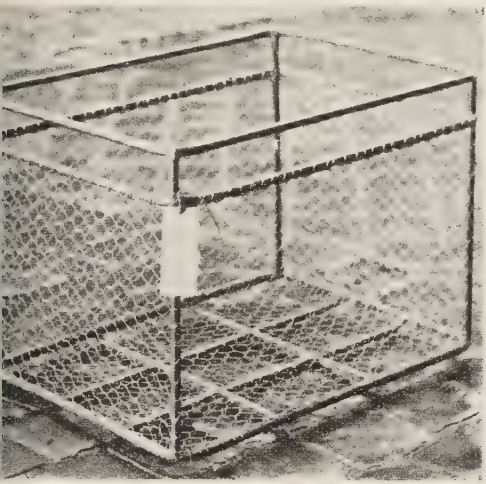


Fig. 9. — Basket containers for post parcels.

on to 30 easel-shaped stands constructed of ordinary rolled sections assembled by electric arc welding.

Special basket containers were introduced for conveying fragile parcels not suitable for bagging. These containers (see fig. 9) consist of a welded framework of light steel sections and tubes, covered with a light iron trellis. Weighing only 6 kgr. (13.2 lb.), they will hold 175 dm^3 (16 cu. ft.)) of parcels, representing an average load of 35 kgr. (77 lb.).

Results of mechanisation.

The mechanical installation at Brussels-Midi station enables about 20 000 parcels to be handled daily. The concentration of parcels at the eleven offices indicated in the sketch map (fig. 3) has been abolished so far as parcels travelling via Brussels-Midi are concerned, the operations at these points being now confined to parcels that do not require to be routed via that station. The concentration area of Brussels-Midi thus covers all the lines shewn by a single or double line in the map referred to .

[625 216]

Light automatic drawgear couplings for the principal European Railways,

by Jacques HENRICOT, Court-Saint-Etienne (Belgium).

The automatic coupling of vehicles has interested railway companies since their inception. The use of automatic couplings in the United States has been compulsory by law for more than 40 years.

The American wagons being equipped at that time with a central buffer coupling, the fitting of the automatic central buffer coupling could be carried out without great difficulty. The coupling adopted was the M.C.B. or A.R.A. automatic coupling.

The lead given by the United States was followed by many countries and now couplings of this type have been adopted by the majority of the railway companies of Central and South America, Central and South Africa, Australia, China, Japan, etc.

Europe alone has not, up to the present, advocated the general use of automatic couplings. The question, however, has not ceased to occupy the attention of those interested in it who, profiting by the experience gained elsewhere, have considered for many years the possibility of introducing an automatic central buffer coupling on the standard-gauge European railway system.

In Europe, however, the introduction of automatic couplings is much more difficult, as the rolling stock is fitted with side buffers. Moreover, the large number of systems and their intercommunication would necessitate unanimous decisions respecting the manner of introduction and the type of coupling to be adopted, and thus complicate the question still further.

In order to solve this problem, the International Union of Railways (U.I.C.) set up, in 1926, an Automatic Coupling Committee which studied particularly the question of the transition from the present screw coupling gear to automatic central buffer coupling. It was considered that this latter would be designed to resist much higher impacts and tractive efforts than the gear now fitted in Europe. After four years of study, this Committee has recommended, in its « Provisional Conditions », the following procedure for transition.

The existing vehicles, for the time being, would retain their side buffers and would be equipped with automatic couplings fixed to the draw hooks and designed to transmit tractive forces only (drawgear couplings). The automatic coupling of rolling stock in this way being obtained from the outset, the second stage of the transition would consist of equipping new vehicles with central buffer couplings capable of being coupled to the automatic drawgear coupling, while continuing to fit side buffers to new stock.

Finally, as soon as the vehicles equipped with automatic drawgear couplings would have been withdrawn from service, and only vehicles fitted with automatic central buffer couplings remained in service, the side buffers would be taken off. Then the conversion to central buffer coupling would be fully accomplished.

Amongst the many schemes for transition proposed up to the present time for Europe, this scheme appears to be the only one which is practicable under the operating conditions of the European railway systems and at the same time it reduces to a minimum the cost involved. However, according to the calculations made by the Coupling Committee of the U.I.C., this scheme would still involve a too high expenditure for it to be easily borne by all the European systems. For this reason the Coupling Committee of the U.I.C. stated on the 20th June, 1936, that « ... from an economic point of view the replacement of the screw coupling gear, as now used, by an automatic central buffer coupling would not be justified ».

One is thus led to consider simply the automatic coupling of stock by means of a coupling transmitting tractive forces only (drawgear coupling) that is to say, similar to that proposed by the U.I.C. during the transition period.

But this drawgear coupling used during the transition period must, according to the recommendation of the U.I.C., allow of its being coupled with the automatic central buffer coupling with which new vehicles would have been subsequently equipped. If this condition is waived, the question is simplified considerably.

In fact, the automatic central buffer coupling must be heavy and bulky, on the one hand to resist the impacts and tractive efforts anticipated in the future, and on the other hand must have a very extended range of action because automatic central buffer couplings which do not couple when vehicles are brought together would lead to serious damage, as the wagons are no longer provided with side buffers.

But if it is essential that the automatic drawgear coupling must couple with such an automatic central buffer coupler, its dimensions must correspond with the latter, and, therefore, it must itself be of a much bulkier and heavier construction than otherwise necessary, which would mean a considerable increase in the price of the drawgear coupling.

It, therefore, follows that a drawgear coupling not intended to couple with the automatic central buffer coupling will be lighter and cheaper.

We can thus conclude that the automatic coupling of vehicles provided with side buffers can be attained by fitting a light drawgear coupling to the draw-hooks.

With regard to new vehicles, the subsequent fitting of central buffing and drawgear can nevertheless be provided for by allowing the necessary space; however, in the meantime, these vehicles would be fitted simply with side buffers and light drawgear couplings. When in years to come there will be in service only vehicles where provision has been made to fit central buffer couplings, the question of adopting a type to suit the conditions of the traffic can be reconsidered freely.

In this manner the introduction of automatic couplings would be carried out at a very reduced cost by means of simple and light drawgear couplings and if, later, the adoption of central buffer couplings appears desirable, the large economy effected by fitting drawgear couplings will facilitate this new course.

* * *

The complex problem of automatic coupling has received the attention of many inventors. Amongst others, the firms specialising in the manufacture of

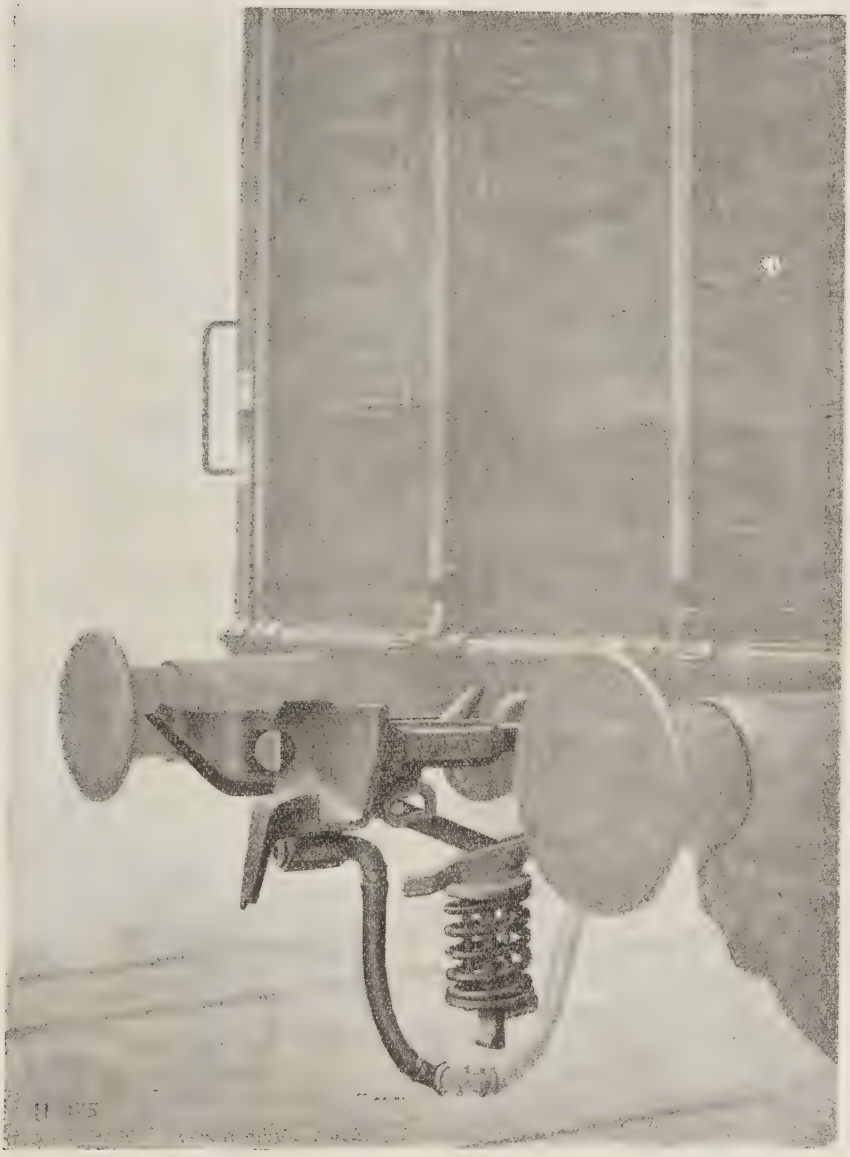


Fig. 1. — Standard wagon with *Compact léger* (Light Compact) drawgear coupling.

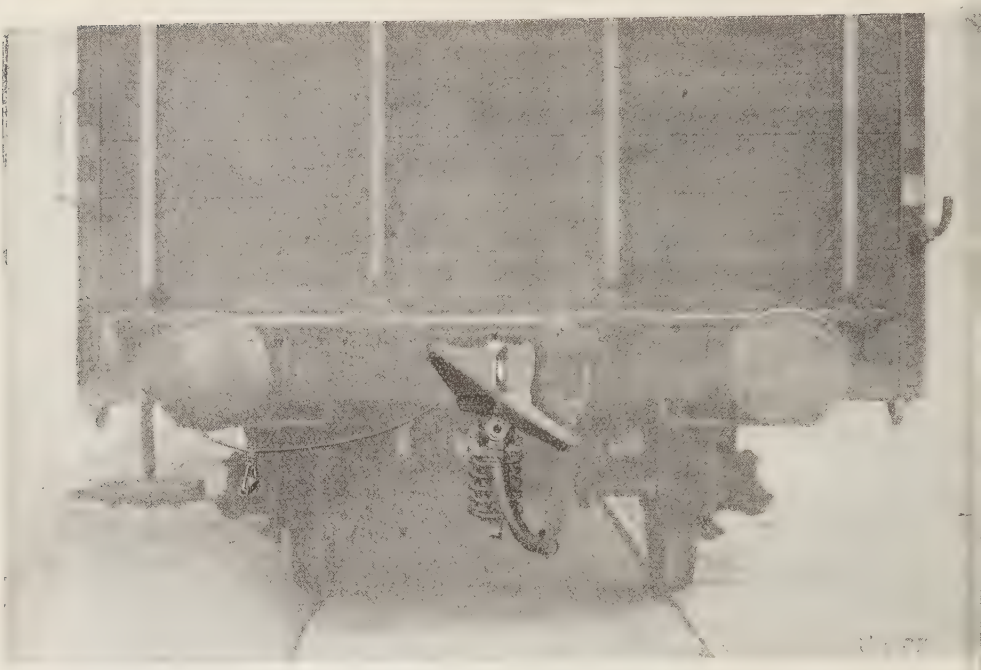


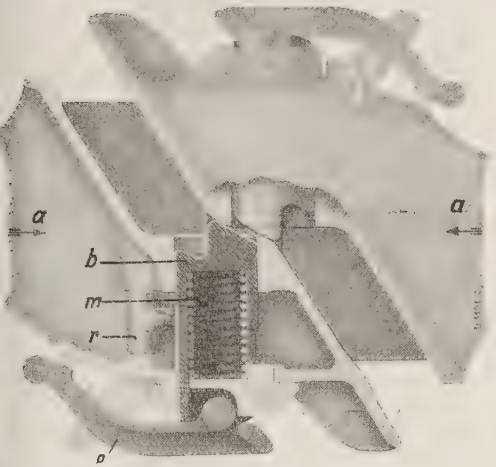
Fig. 2. — Automatic *Compact léger* drawgear coupling. End view.

couplings which have developed and manufactured Boirault, Henricot and Union (Kürtössy) couplings, have associated in making a new coupling called « *Attelage Compact* » (Compact Coupling), which agrees with all the recommendations of the U.I.C. both as a transition type of coupling and as a central buffer coupling. Moreover, these firms have designed a type of drawgear coupling called « *Compact léger* » (Light Compact) which is of a particularly simple and light construction, (see figs. 1 and 2); it agrees with all the conditions set forth by the Coupling Committee of the U.I.C. except that which calls for the possibility of connecting with the heavy central buffer coupling.

But this light coupling can be coupled to a light central buffer coupling designed to resist only the impacts and tractive efforts which are found in actual service It would be possible, therefore, to coupl vehicles with central buffers to thos fitted with « Compact léger » couplings

The coupling head is made up entirel of guiding surfaces profiled to ensur the engagement of two couplings whic often come together out of alignment ver tically and horizontally. This engage ment becomes tighter as the coupling approach each other and ends by makin a rigid connection between the two coup ings which are then locked in position b the mutual interlocking of two bolts. Th couplings thus locked can be disengage only by operating the release cord. A the essential parts, proportioned to with stand the forces, are made the minimur weight compatible with security; ing nious arrangements guard against th possibility of mud or dust checking th

Fig. 3. — Action of the automatic *Compact léger* coupling.



A. During coupling.

B. Apparatus coupled.

C. Apparatus uncoupled (but not separated).

proper working. The apparatus is very simple and apart from the suspension necessitates only two component parts for the transmission of the tractive forces, (fig. 3) : the body *a*, forming a cavity for the locking device, and the wedge bolt *b*. Its spring *m* is hidden in the interior of the bolt; the release cord operates the release lever *p*, the catch *r* automatically holds the bolt in the open position until the vehicles are separated.

The precision of the guiding arrangement of the apparatus enables the brake pipes to be readily coupled, thus eliminating the necessity for staff to go between the buffers.

The weight of the *Compact léger* drawgear coupling scarcely exceeds that of the screw coupling and the emergency coupling. The additional weight, which is only 25 kgr. (55 lb.), allows of the replacement of the screw coupling by a completely automatic coupling : the *Compact léger* coupling.

Fig. 4. — Lowering the apparatus.

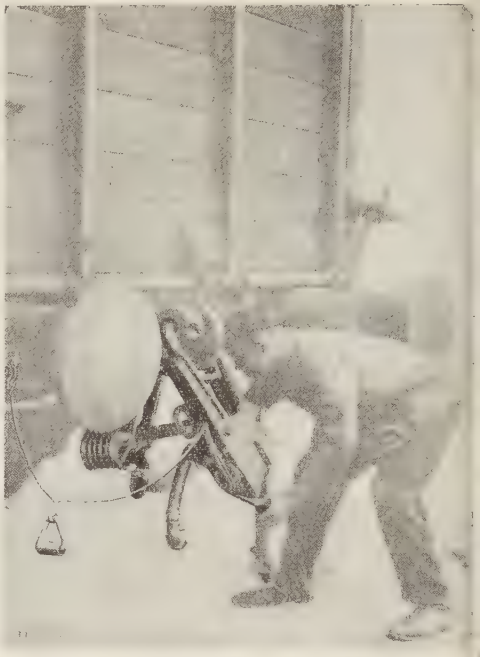


Fig. 5. — Replacing in service position.

It would be possible for the different countries to adopt automatic coupling independently of one another. As a matter of fact, this *Compact léger* coupling can be quickly and easily lowered under a vehicle by one man (fig. 4), the draw hook being thus freed to allow of coupling to stock not fitted with an automatic coupling. Also the replacement of the coupling can be easily carried out by one man (fig. 5).

Because of the low cost of the proposed coupling, the simplicity of its arrangement and the ease with which it can be fitted to vehicles, without modifications to them, it is evident that the introduction of the automatic drawgear coupling only requires a small fraction of the expenditure considered up to the present.

Further, it can be said that most of the economic advantages which would result from the adoption of an automatic central buffer coupling would be obtained by the introduction of the *Compact léger* drawgear coupling.

Taking into account only the economies anticipated by the U.I.C. in its *Bulletin*, it would appear that the cost of purchasing these couplings, including the interest on capital expended, could be redeemed in less than 6 years from the date the *Compact* coupling was put into service.

[625. 143 (47. 42)]

An investigation into the present condition of rails laid before the War on the main lines of the Esthonian State Railways,

by P. KANDAOUROFF, Engineer, Paris.

Some 400 km. (250 miles), or 52 %, of the 772 km. (479.7 miles) of main line of the Esthonian State Railways, which were constructed to the Russian gauge of 1.525 m. (5 ft.), are laid with rails dating from before the War and known as the Russian rail sections *2bis*, 5 and *11*, weighing 29.83, 32.69 and 32.92 kgr. per metre (60.1, 65.9, and 66.3 lb. per yard) respectively.

Some of these rails are as much as 66 years old; the newest were rolled in 1904, after which these types of rail were abandoned. They were not always new when put down on sections of line forming part of what is now the Esthonian railway system, but were usually laid first on the Russian lines, where they were replaced in due course by others of heavier type.

No information is available, even approximate, concerning the amount of traffic that has passed over them. Their condition is not only a result of long years of service, but of 6 years defective maintenance from 1914 to 1920, covering the war period, increased to some extent by the use of worn fastenings, which often did not fit the rails to which they were attached. Speeds have also increased in the course of time and the average axle load has grown from 12 to 16 tonnes (11.8 to 15.7 Engl. tons), so that present conditions are far more difficult than were those obtaining when the rails were laid. All these circumstances have combined to increase the number of rail breakages, which is still rising and threatens to become serious in the near future.

In 1931 there were 48 rail breakages in the 772 km. (479.7 miles) of track, but in 1935 there were 148, or just over 3 times as many in 5 years. About 80 % of the breakages were in rails of the *2bis*, 5 and *11* types, and as their total length is about equal to that of all other types on the Esthonian railways, it can be said that the former break 4 times as often as the latter, which chiefly consist of the newer Russian type III*a* rails, weighing 33.48 kgr. per metre (67.6 lb. per yard) and are not more than 32 years old. On the other hand, there are some fairly long sections of line where traffic is light, on which rails of the *2bis* type were laid new in 1905, and where they are in much better condition. Breakages are naturally far less frequent in this case and hence the proportion of breakages of the *2bis* rails, compared with the 5 and *11* types, is much smaller.

The number of breakages of the latter types, however, increased markedly in 1935. Thus there were on the average 45 breakages of the type 5 rails per 100 km. (62 miles) of track during the years 1931 to 1934, but in 1935 the figure rose to 103. Similar figures apply to the type *11* rail, the breakages being 10 per 100

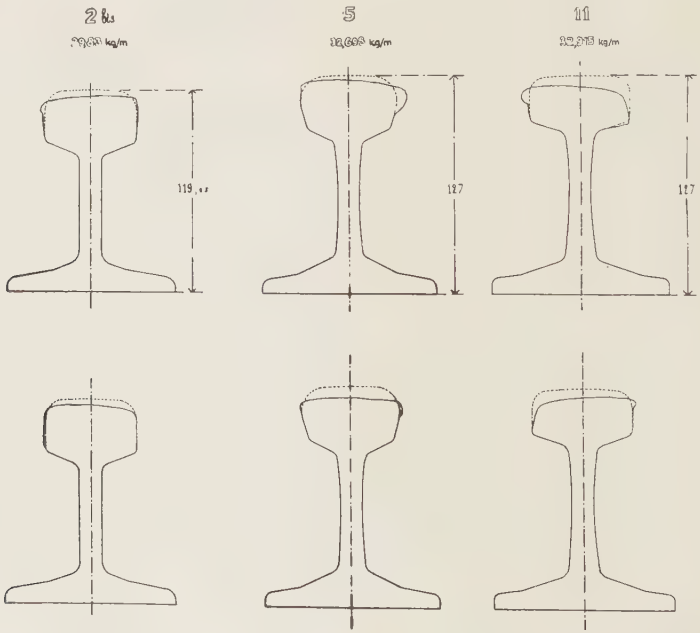
km. for the years 1931 to 1934 and 22 in 1935. The number was thus 2 1/4 times as great in that year, although traffic and weather conditions were practically the same as in the previous years mentioned. 105 broken type 5 rails for 100 km. of line is $\frac{1}{250}$th of the total number in use and is a very high figure. In many countries regulations require the elimination of rails of a given cross section, origin and age if 1 per 1 000 of them break annually.

* * *

In order to ascertain more accurately the degree of reliability for further main-line service of the type *2bis*, *5* and *11* rails, rolled between the years 1870 and 1903, the Management of the Esthonian State Railways undertook a thorough examination of them in 1934 and 1935, making careful cross-sectional measurements on the ground and carrying out tests, particularly impact tests, at a testing station. Altogether some 1 672 rail sections were drawn by means of the profilograph, taken approximately in the centre of rail lengths lying in main lines. The varying dimensions at the end and middle were also measured, which enabled the total rail wear and the batter at the rail ends to be estimated. The figures in this article show the extent of this wear. It was found that the *average* wear, not that at the rail ends, of the various classes of rail was :

For type *2 bis* rails . 2.4 mm. (0.0945 in.)
For type 5 rails . . 2.3 mm. (0.0905 in.) and
For type *11* rails . . 1.9 mm. (0.0748 in.)

The wear at the rail ends was naturally more considerable and varied, but pro-



Scale : 1.4.

trusions and side wear were only found here and there. There were few rails in which the wear amounted to as much as 5 mm. (0.197 in.) the figures being :

For type *2 bis* rails, 4 per cent of all rails,
For type *5* rails, 1.5 per cent of all rails,
For type *11* rails, 4 per cent of all rails,

showing that generally speaking, from the point of view of wear, the rails are still fit for service. It should be pointed out that the rails laid under the Russian administration were put down without regard to their origin, date of rolling, position of the rolling marks or amount of wear.

* * *

To investigate the quality of the rail material drop tests were made first, using a cast iron tup weighing 494 kgr. (1 089 lb.). This had a steel impact rib, in accordance with Russian specifications, and was allowed to fall on a piece of rail 1.30 m. (4 ft. 3 1/4 in.) long, carried on 2 supports of prescribed form placed 1.00 m. (3 ft. 3 3/8 in.) apart, resting on a concrete baseblock weighing 16.5 tonnes (16.2 Engl. tons). The impact tests were made on rails removed from the main line on a distance of 10 km. (6.2 miles) and supplied by various rolling mills. It was found that in a length of 368 km. (228.7 miles) there were rails from 12 different makers.

Two methods of making these drop tests were followed : The older, prescribed in the « Technical specifications covering the Supply of Steel Rails » issued by the Russian Ministry of Transport in 1917, and the newer, introduced in Esthonia in 1929 and generally in accord with the requirements of the railways in Western Europe. The height through which the tup is allowed to fall is determined by different constants for the two methods, depending on the weight of the tup, 492 or 1 000 kgr. (1 084 or 2 204 lb.), the distance between the test piece supports, 1 067 or 1 000 mm. (3 ft. 6 in. or 3 ft. 3 3/8 in.), and the length of the piece, 1 524 or 1 300 mm. (5 ft. or 4 ft. 3 1/4 in.) and, in addition, varying in proportion to the reduction in the moment of inertia of the rail and the distance from the outer layers of the material to the neutral axis of the rail, as varied by wear. In this way the amount of load necessary in each case to effect the prescribed deformation was arrived at.

It is as well to point out here an important difference between the Russian and Esthonian specifications. While the former prescribes that the test piece must twice withstand a mechanical work equal to the product of the weight of the tup and the length of fall without the deflection produced by the first blow exceeding 75 mm. (3 in.), the latter stipulates that a deflection of 100 mm. (4 in.) is to be obtained with any given number of blows. To achieve this object the Esthonian specification only lays down the height of fall for the first blow. This fall, and the deflection produced thereby, are thus greater under the conditions set forth in the Esthonian than in the Russian requirements.

It is clear from the above conditions that the total load tending to deform the rail, i.e. the sum of the products of the weight of tup and the height of fall necessary to obtain the deflection of 100 mm. (4 in.) stipulated in the Esthonian specification, is really as a rule less than that called for by the Russian requirements.

It follows, therefore, that to meet the

latter a tougher and less elastic material, with higher tensile qualities, must be used than is required to meet the Esthonian specification, which calls for less tensile strength (less total effort to effect the same deformation) but a greater capacity for deflection and greater toughness. This circumstance is in direct conflict with the remaining conditions in the two specifications, for the Russian stipulates an ultimate tensile strength of only 65 kgr. per mm^2 (41.27 Engl. tons per sq. in.) and the Esthonian one of 70 kgr. per mm^2 (44.44 Engl. tons per sq. in.).

It should here be observed that if the rail does not break under the drop test, the tup, owing to the elasticity of the rail, rebounds to a certain extent — about 0.50 m. (20 in.) after the initial blow — and strikes the rail several times again. The additional effect due to these blows, which gradually die out, may, to all intents and purposes, be neglected in practice, although it can at times be appreciable.

The test pieces were cut from parts of the rail where no fishplates had been attached, in order to eliminate any influence due to wear on the rail head near the rail joints and those portions touched by the fishplates, thus excluding any alteration to the structure of the metal caused thereby. From each rail examined, 5 pieces were cut, the first for use in the testing laboratory, the other four being used for drop tests, under the Russian and the Esthonian specifications alternately, at a temperature of 3 to 12° C. (37.4 to 53.6° F.) Rails supplied by 10 Russian, 3 English and 1 French rolling mills were dealt with, with the following results :

25 % of all rails met the tests of both specifications,

14 % complied with the Esthonian specification only,

33 % complied with neither.

In 28 % of the tests the results were indefinite, that is to say one piece of rail complied with one of the specifications and the other failed to do so. There were no rails which complied with the Russian specification and not the Esthonian, which indicates that the Western European requirements are less severe than the Russian, although the Esthonian specification presupposes stronger material than the Russian, as is confirmed by the above mentioned inconsistency between them.

Unfortunately it was not possible to draw conclusions concerning the properties of the rails pointing definitely at any particular rolling mill. Many rails from one supplier passed all tests while others failed, due no doubt to the fact that they came from different heats and were supplied over a considerable space of time. Conditions on different parts of the line also varied a great deal.

The results of the drop test did not agree with the behaviour of the rails in service. For example, some type 5 rails supplied by a well known rolling mill showed good results under that test, but nevertheless figures proved them to be the worst from the point of view of breakages in service. The general conclusion might therefore be drawn that only some 25 % of the total number of rails on the main lines can be left there and that 33 % are unfit for the service now obtaining. It is impossible, however, to select rails according to their origin or the year in which they were rolled. Comparison between drop tests made on different pieces of the same rail also shows that they behave better the farther they are from the ends of the rail, indicating

hat the alteration in the qualities of the naterial, or in its fatigue, under blows t the rail joints, is not equally distributed along the rail.

In addition to the drop tests the following tests were made at the National Testing Laboratory, viz. : — for ultimate ensile strength, determination of the limit of proportional elongation, the modulus of elasticity, specific elongation nd contraction, on test pieces cut from he head, web and foot of each rail examined. Brinell hardness tests were also nade at 12 places on the cross section, s well as a metallographic examination f the macrostructure of the steel in the ail section and of its microstructure in places.

The following conclusions were arrived at :

(1) The ultimate tensile strength of est pieces taken from the same piece of ail varies considerably, there being differences as great as 5 kgr. per mm² (3.17 Engl. tons per sq. in.), or some 10 %, ccording to the part of the rail from vhich the test piece was taken. In 70 % f the cases the lowest tensile strength vas found in pieces taken from the web f the rail.

The ultimate strength varied between 2 and 49 kgr. per mm² (58.4 and 31.1 ngl. tons per sq. in.), being in 40 % of ne cases lower than the figure in the sthonian specification, 70 kgr. per mm² 44.44 Engl. tons per sq. in.), and in 0 % being less than the Russian requirement of 65 kgr. per mm² (41.3 Engl. ons per sq. in.).

(2) Although the results were generally satisfactory as regards the limit of roportional elongation, nevertheless here were considerable variations disosed, from 23 to 53 kgr. per mm² (14.6 to 33.65 Engl. tons per sq. in.), so that in one and the same rail section its variation was as much as 100 : 163. It is noteworthy that the limit figure was much lower for the head of the rail than for the web or foot. In 90 % of the cases it was insufficient to meet the lower limit of 32 kgr. per mm² (20.32 Engl. tons per sq. in.) laid down in the Esthonian specification and in 20 % did not satisfy the lower Russian limit of 25 kgr. per mm² (15.87 Engl. tons per sq. in.). In one case the result was as low as 22.6 kgr. per mm² (14.35 Engl. tons per sq. in.). The variation of this characteristic in the same rail section is particularly striking, amounting to a proportion of 100 : 186.

(3) Elongation was in every case satisfactory and within the limits of 12 and 28 %, the Esthonian specification calling for a minimum of 10 %, while variations found in the same rail section were in the proportion of 100 : 150. The least amount of elongation was generally found in pieces taken from the head of the rail.

(4) The Brinell hardness tests everywhere revealed greater hardness in the head than in the foot of the rail, as was to be expected from the lesser elongation in the head. The Brinell number varied between 124 and 187 in the same rail section, or a proportion of 100 : 150, confirmed by the above mentioned irregularity in the character of the metal. Microscopical examinations have revealed such differences that it is hard to believe that the photographs can be of the same rail section, variations being especially noticeable in the centre of the head and web of the rail, where the metal generally exhibits a special microstructure. A somewhat considerable want of homogeneity appeared in all the photographs,

confirmed by the various tensile and hardness tests.

Although the photographs revealed considerable segregation of the metal in some of the rails, the mechanical tests showed such rails not to be among the worst, the drop test generally giving good results.

A comparison between the results of drop and tensile tests and microscopical examinations shows it to be impossible to harmonise them. Rails with defective micro- and macrostructure gave excellent results under drop and tensile tests, while others possessing good mechanical properties failed under the drop test. Some of the above-mentioned rails, which came from a particular rolling mill gave good results under the drop test, but had the greatest number of fractures in service; they showed up poorly, however, under mechanical and metallographic examination. The latter, therefore, confirmed the results obtained in service, but the drop tests did not, *and yet this is everywhere regarded as being a particularly valuable test.*

These careful investigations by the Esthonian State Railways lead to the conclusion that rails in a main line, which have reached a certain age and suffered a certain degree of wear, depending on the speed and size of the loads passing over them, are to be looked upon with suspicion and should be removed, in order not to impair the safety of the traffic. This is now being rapidly done on the Esthonian system. Nevertheless it is impossible to establish any accurate relationship between the various characteristics of the metal, ascertained by any form of laboratory test, and the ability to withstand fracture in service, which can only be revealed by statistics of breakages.

It is true that rails so taken out of service can still be used for a long time in sidings or on secondary running lines, and it is the intention to do this in Esthonia. This is of importance in colonies and overseas in places where the railway system is still but little developed. In such cases it would be advantageous at first to use short, light, old rails, still fit for service, and replace them as traffic develops with new, longer and heavier ones. The old ones can then be used for the extension of secondary lines and branches, seeing that new rails put down during the building of a line unavoidably receive a certain amount of damage. This was the case, for instance, on the Trans-Siberian line, where rails weighing 22.5 kgr. per metre (40.35 lb. per yard) were laid to begin with. They were later replaced by heavier ones and at the present time nearly all the original rails are to be found on various Siberian lines of lesser importance.

[62.(01 & 625. 13]

Finding weak spots in bridges,

by Rudolf K. BERNHARD,
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(Engineering News-Record.)

As an advance on static and moving load tests, European engineers now apply induced vibration to bridges as a means of locating defects, especially of welded structures, before and while they are in service.

Bridge testing to give assurance of safety has been going through a significant change in European countries within the last few years. For a long time past it was the rule in European countries to test a newly completed bridge by subjecting it to a test load before putting it into service, a practice not usually followed in America. A few years ago, however, higher speeds of railway trains became common, and at the same time welding found wide use to replace rivets in the connection of members and parts of members of bridges. Static testing was felt to be no longer adequate under these conditions, and dynamic testing was resorted to. The characteristic feature of this form of testing as now practised is placing the bridge under induced vibration by means of an oscillator, and measuring its behavior under such action.

Up to about ten years ago the common practice in the case of a just completed railway bridge was to run a train on the bridge, representing as closely as possible the maximum locomotive and train loading for which the bridge was designed, and to measure the amount by which the bridge deflected under this load. In the case of a highway bridge, loaded trucks or steam rollers were used. By comparing the measured deflection with that calculated from the dimensions of the bridge and members certain conclusions could be drawn. For example, if a serious irregularity of the deflection curve was observed, or if the deflection was larger than that predicted from calculation, defects were looked for.

This method unfortunately gave no warning of a weak detail, since a small local weakness would not change the deflection perceptibly even though it endangered the security of the bridge. Also, the comparison with calculated deflection was vitiated by the influence of stringer participation in chord stress and by secondary stresses. When welding came into use the possibilities of trouble from local defects in the welding were recognized as much more serious than the corresponding risks in a riveted structure, and attention was turned to finding more certain methods of detecting bridge weaknesses.

The introduction of welding on a large scale in European continental bridge practice, roughly ten years ago, coincided with a trend toward higher railway speeds and heavier loads. At present Germany alone has several hundred welded bridges, including both railway and highway bridges, the former mainly plate-girder structures while many of the highway bridges are welded trusses. Due to the nature of welding, the danger of local connection defects is distinctly greater than in riveted bridges. If, for example, the welding at any one point in the bridge is defective by

reason of imperfect adhesion, slag inclusion, burned metal, or by high shrinkage stress, a crack is likely to develop under load, which in due course would tend to cause failure of the bridge. Therefore the shortcomings of static load testing received close study, with a view to finding means of assuring safety by some more positive method.

Dynamic testing introduced.

It was concluded that dynamic testing should supplant static testing, and many tests were made under moving loads, both deflections and stresses being measured. This procedure was nothing more than the application to regular service use of methods long previously applied in experimental research work on bridges, notably in the work of the British bridge impact committee. (1928, Report of the Bridge Stress Committee, London, England).

In 1928 dynamic testing was further developed by the experimental use of oscillators to vibrate the bridge. The behavior of the bridge under such vibration proved to be a much more sensitive indication of weaknesses in the structure than either moving-load or static tests. Within two years such vibration testing had virtually become established as a necessary supplement in the verification of bridge strength, and came into regular use by the German State Railways for welded bridges. Since then it has also found extensive use in Russia, Italy, Switzerland and France, though more for research purposes than for acceptance testing.

Value of vibration tests.

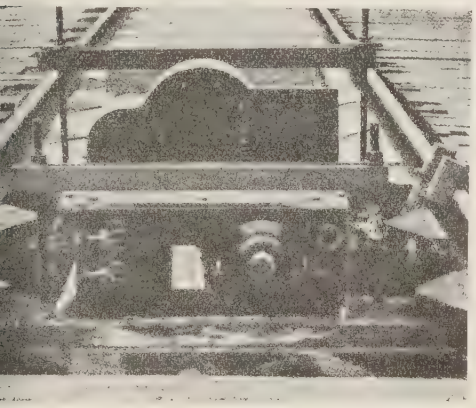
Stated in simplest terms, the value of the vibration test lies in the fact that it measures the elasticity of the bridge, or its associated property, the degree of damping. So long as all parts of the structure are in sound condition and are not overstressed, the bridge wil have a relatively high elasticity and a low damping; but points of overstress and still more so loose joints or cracke welds, will absorb a certain amount o power during the vibration, which wil increase the damping coefficient. Whe excessive damping points to the exist ence of some local defect of structure including foundations, a search for th weakness can be made by thorough in spection, measurements of local stresses and if necessary by X-ray examinatio of the welds.

The method can be applied to structure of any size, new or old, fi nished or unfinished, or to any compo nent part of the bridge. Static or mov ing loads may be combined with the vi bration. It affords a means of compar ing different types of structures, in cluding concrete bridges. The many vi bration tests carried out by the Germa and Swiss railway systems since 192 on structures varying widely in size, ma terial and type, have proved the simpli city, economy and usefulness of the m thod.

A bridge may vibrate in any direc tion when a moving load crosses it. I railway bridges such vibrations ar caused by worn or flat car wheels o the moving load, poorly laid track, an unbalanced reciprocating parts of th locomotive. In highway bridges uneve road surfaces cause vibrations. Thes vibrations may result from single sho blows or longer sinusoidal impulses an are more often irregular than regula Studying the effects of the vibration caused by the moving loads is a com plex problem and presents many diff culties.

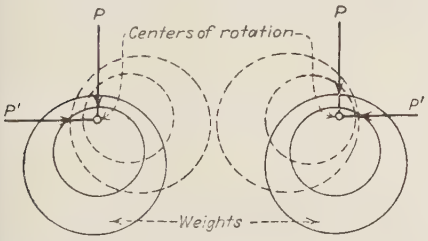
Induced vibration of the bridge by a oscillator does not solve these difficu ties by reproducing the vibrations cau ed by moving loads, but rather subject the bridge to a regular vibration whic aids in locating structural weaknesse The oscillator, (fig. 1) consisting of tw

eccentric weights rotated in opposite directions (fig. 2), applies regular sinusoidal forces to the bridge. Both the frequency and amplitude of the resulting oscillation can be varied by varying the speed and eccentricity of the rotating weights. The power input to the



. 1. — In testing a bridge by vibration, n oscillator is attached to the structure nd driven at a speed controlled by the heostat (in foreground).

oscillator at the resonance frequency of the bridge as well as the power input throughout a range of frequencies is an index of the structure's elasticity and its dynamic characteristics. If structural weaknesses do exist, further supplementary investigations, such as measurement of deflections with a deflectometer and measurement of stresses with extensometers, are resorted to.



ig. 2. — In a bridge oscillator two eccentric masses rotating in opposite directions set up sinusoidal forces P, the horizontal forces P' neutralizing each other.

Equipment used in testing.

The oscillator, which is similar to that originally used by the British committee, is motor-driven and is equipped with a rheostat speed control, a speed indicator, and a wattmeter. Oscillators can be built to operate at frequencies between 1 and 100 cycles per second and will provide impulses between 2 lb. and 20 tons. In testing a bridge, the magnitude of the oscillations is amplified by the induced vibration of the bridge itself. This amplification increases as the frequency of the oscillator approaches the natural frequency of the bridge and in some bridges may be as much as 40 times.

A special test car, developed by the German railroads, transports the entire equipment necessary for a bridge test by artificial vibration. The car contains photographic dark room, machine shop, generator, and living quarters for the test personnel, making test results immediately available in the field where investigations to locate any structural weakness can proceed immediately under favorable conditions.

In locating structural weaknesses by stress measurements the German railroads adopted as their standard instrument the carbon-stack extensometer (fig. 3). The variation of stress in a bridge member as the bridge vibrates is recorded photographically by an oscillograph connected to the extensometer. Eight or more extensometers can be connected to one oscillograph, permitting eight or more different parts of the bridge to be observed simultaneously.

Testing a bridge.

In testing a bridge by the induced vibration method, the oscillator is attached to the bridge (fig. 1) in a position

which will produce the desired vibration. Particular care is taken to avoid such power losses as those caused by energy dissipation through the foundations. The weights of the oscillator are

Fig. 3. — Stresses set up by the oscillator are measured by an extensometer and recorded by an oscillograph. The carbon-stack extensometer is here shown attached to the flange of a bridge girder.

then rotated, starting with low frequency and increasing the frequency by steps. The power demand of the oscillator at the various frequencies is measured and a power input-frequency curve (fig. 4) is drawn for the bridge being tested.

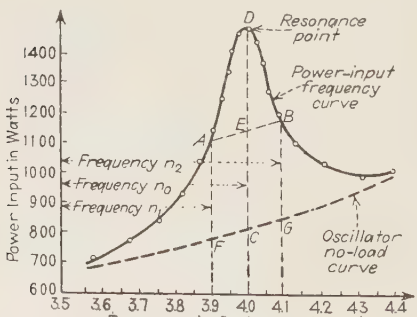


Fig. 4. — The amplifying factor and the damping coefficient computed from the power input frequency curve of the oscillator indicate whether there are weaknesses in the structure. This diagram is for a 310-ft. single track truss bridge at Vlotho, Germany.

Maximum power demand of the oscillator occurs at the resonance frequency of the structure. Power demand data of tests, made on the oscillator when it is not attached to the bridge and is running free, are used to plot the no-load curve of the oscillator (fig. 4). From the power input-frequency curves, for the oscillator vibrating the bridge and also for the oscillator running free, the amplification of the bridge oscillations is determined.

This amplifying factor is computed by constructing line AB (fig. 4) so that it bisects CD at point E and makes AF, EC, and BG all equal. The amplifying factor depends upon the relation of the three frequencies n_1, n_2, and n_0 determined by the above construction of line AB, and is equal to

$$\frac{n_0}{n_1 - n_2}$$

The associated property of amplification, the damping coefficient, is equal to π times the reciprocal of the amplification factor.

Tests conducted by the German State Railways indicated that the usual welded structural steel bridge with all parts in sound condition, no loose joints, and no cracked welds, has a damping coefficient between 0.02 and 0.2. A riveted structural steel bridge in similar condition has a damping coefficient between 0.05 and 0.5.

Discovering weaknesses.

If a vibration test of a bridge shows a damping coefficient greater than the range for that particular bridge it points to a power loss at some structural weakness, such as a loose joint, an overstressed detail or a cracked joint. A search for the structural weakness is conduct-

ed by thorough systematic inspection of all members and connections, measuring local stresses with extensometers, or in some cases by X-ray examination of the welds.

Possibilities of resonance.

The test method of artificially vibrating a structure can also be used in studying another important phenomenon in the structural field, namely, resonance between vibrations of the live load and the bridge. The situation is analogous to that of a large mass (the bridge) connected to rigid abutments by springs, with a smaller mass (the live load) connected to the large mass by a spring. The characteristics of the individual vibrations of each of the two masses markedly affect the resulting joint vibration of the system. For example, a locomotive situated in the middle of a single-track railway bridge of the same weight and of ordinary span can produce a resulting vibration frequency which is seven-tenths of the natural frequency of the unloaded bridge. Studies of the natural frequencies of bridges and moving loads for various types of vibrations have aided in determining the conditions under which resonance is of importance.

The oscillation of two coupled systems which have approximately coinciding natural frequencies creates what is known as beat. The amplitudes of the two masses increase one after another to a maximum and then decrease to a minimum. The time interval between two successive maxima is the period of the beat.

Particularly interesting is that oscillation in which a complete energy transfer occurs. In such a case, one mass vibrates and then comes completely to

Natural frequencies, damping coefficient, and amplifying factor of railway and highway moving loads.

	Load.			Natural frequency in cycles per second			Damping coefficient.	Amplifying factor.
	Type.	Description.	Condition of loading.	Vertical.	Horizontal in direction of track.	Torsion oblique to track.		
Railway loads	Locomotive	Medium speed engine, on 5 axles	Boiler full, tank empty. Total weight = 62 tons	Under 2.0	6.6	8.7	0.51 0.71 0.90	6.2 4.4 3.5
	Locomotive	High speed engine, on 6 axles	Full weight = 113.5 tons	Under 2.0	5.4	3.7	0.29 0.77	6.2 3.5
	Passenger coach	...	Empty	Under 2.0	2.6	...	0.36	8.7
	Heavy freight car	...	Loaded with 55 tons of coal	6.3	5.9	5.8	0.37 0.53 0.33	8.5 5.9 9.5
Highway loads	Automobile	30-ton truck with high-pressure tires	Empty	3.5	...	1.9	1.17 0.79	2.7 4.0
	Automobile	17-ton truck with low-pressure tires	11-ton load	4.1	2.1	...	0.34 0.50	9.2 6.2

rest. While this is occurring the second mass reaches its maximum movement. Then the second mass comes to rest while the first mass is again reaching its maximum. Such a complete transfer of energy visualizes the approximate coincidence of the natural frequencies of both oscillation systems in an uncoupled condition. This complete transfer of energy cannot occur when the oscillating bodies possess damping characteristics.

Natural frequencies of bridges.

Numerous bridge tests (« Impact in steel railway bridges of simple span » by J. B. Hunley, A. R. E. A., Vol. 37, October, 1935) have provided the following approximate formulas for the natural frequencies of bridges subjected to the three main forms of oscillation :

For vertical oscillation, $n = \frac{800}{L}$

For horizontal oscillation, $n = \frac{370}{L}$

For torsional oscillation, $n = \frac{935}{L}$

In these formulas L is the bridge span in feet and n is the natural frequency of the bridge in cycles per second.

For ordinary bridge spans vertical oscillations range from 3 to 16 cycles per second, horizontal oscillations are slower and range from 2 to 14 cycles per second, while torsional oscillations are faster and range from 5 to 17 cycles per second.

Natural frequencies of loads.

The natural vibration frequencies of locomotives, passenger coaches, freight cars and automobiles are affected by such factors as amount of the load carried, type and size of springs, and number of axles. For example, that part of the moving load supported on the springs of the car may move in harmony with the wheel trucks or its movement may be quickly damped by the springs. Several typical railroad and highway vehicles were vibrated separately when standing on solid foundations and their natural vibration characteristics were determined. The data of these tests are tabulated in the above table.

The low natural frequencies of locomotives and passenger coaches and the high natural frequencies of heavy freight cars both fall outside the usual range of natural frequencies of bridges but the natural frequencies of ordinary freight cars and automobiles are within the usual range, thus tending to give rise to resonance. All the live loads tested possessed a greater natural damping effect than has been observed on any bridge, however, and so usually do not amplify the magnitude of the bridge's oscillation.

Conclusions.

If only the vertical oscillations of the bridge and its moving load are considered, the following conclusions can be drawn : Resonance is a distinct possibility in long-span railroad bridges of low natural frequency under heavy freight car loads, but natural damping normally prevents high impact. In medium-span railroad bridges, which have a relatively higher natural frequency, resonance with the moving loads tested seems improbable, but unbalanced reciprocating weights of a locomotive may produce considerable oscillation. In railroad bridges of short span, and hence high natural frequency, appreciable resonance cannot occur because the moving load oscillates too slowly and unbalanced reciprocating parts of a locomotive cross the bridge in a very short interval of time.

In highway bridges the occurrence of resonance seems improbable under normal conditions for there are no unbalanced reciprocating parts, and the impulses caused by an uneven road surface are irregular. Road irregularities

however, can cause that part of the vehicle above its springs to vibrate considerably, and if this vibration is in harmony with that of the bridge resonance may result.

The tires of highway vehicles are especially favorable for a transfer of energy from the bridge and the setting up of the beat phenomena mentioned above. In other respects the division of structures into three groups of large, medium, and small spans, as was made for railway bridges, applies in the same way.

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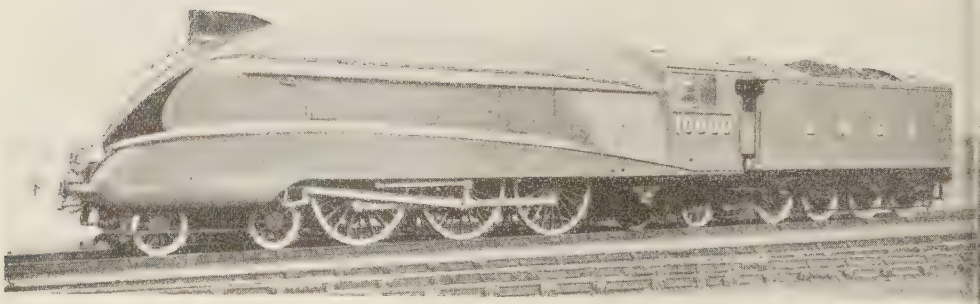
[621. 132.3 (.42)]

RECENT DEVELOPMENTS
IN RAILWAY PRACTICE.

Rebuilding of London and North Eastern Railway engine No. 10 000.



Engine No. 10 000, which was originally built as a four-cylinder compound engine at the Darlington Works of the London and North Eastern Railway, has recently been converted at the Doncaster Works to a three-cylinder simple engine.

Originally this engine was fitted with a Yarrow-Gresley water tube boiler carrying a working pressure of 450 lb. per square inch, and was the first 4-6-4 express passenger engine to be built on the English Railways (1). It was located in the North Eastern Area and was principally engaged in the running of express trains between York and Edinburgh. On occasions it worked the non-stop train from King's Cross to Edinburgh, and proved to be a powerful, fast and free running engine.

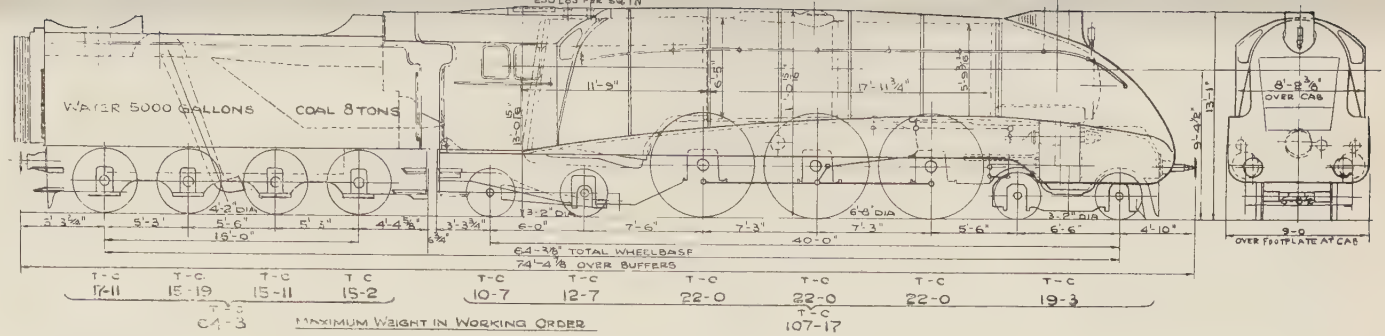
Notwithstanding the use of high pressure, coupled with compounding, the engine did not prove economical in coal consumption, and as it was burning considerably more coal than « Pacific » engines of the standard type, it was decided to substitute a boiler of the type used on the 2-8-2 « Cock o' the North » class engines (2), but with a pressure of 250 lb per square inch.

Whilst the tractive power of the engine as converted is the highest of any six-wheeled coupled express passenger engine on the British Railways, it is not as great as that of the engines of the « Cock o' the North » type.

The boiler barrel has a maximum diameter of 6' 5" and is 17' 11 3/4" between tubeplates. The firebox has a grate area of 50 sq. ft. Two 3 1/2" diameter Ro-

(1) See *Bulletin of the Railway Congress* January 1930, p. 369.

(2) See *Bulletin of the Railway Congress* November 1934, p. 1218.



Leading dimensions and ratios.

Grate:

Length on slope	7' 2"
Width	6' 11 3/4"
Grate area	50 sq. ft.

Firebox:

Height of crown above foundation ring: front	6' 8 13/16"
Height of crown above foundation ring: back	6' 0 5/16"
Interior length at top	10' 2 3/4"
Interior width at boiler centre	5' 4 1/2"
Thickness of copper plates: sides and back	9/16"
Thickness of copper plates: tubeplate	9/16" and 1 1/4"

Boiler:

Outside length of firebox, overall	11' 9"
Outside length of firebox at bottom	7' 11 1/4"
Outside width of firebox at bottom	7' 9"
Maximum diam. of barrel	6' 5"
Thickness of barrel plates	7/8" and 13/16"
Thickness of wrapper plates	9/16"

Tubes, small:

Material	Steel.
Number	121
Diameter outside	2 1/4"
Thickness	10 I. W. G.

Tubes, superheater flues:

Number	43
Diameter outside	5 1/4"
Thickness	5/32"
Length between tubeplates	17' 11 3/4"

Heating surfaces:

Firebox	252 sq. ft.
Tubes	1 281.4 sq. ft.
Flues	1 063.7 sq. ft.
Total evaporative	2 597.6 sq. ft.

Superheater:

Number of elements	43
Diameter inside	1.244"
Heating surface	748.9 sq. ft.
Total heating surface	3 346.5 sq. ft.

Two Ross pop safety valves	3 1/2" diam.
Working pressure	250 lb. p. sq. in.

Axles:

	Dia.	Len.
Journals, pony	6 1/2"	× 11"
— coupled wheels	9 1/2"	× 11"
Trailing	6"	× 11"
Bissel	6 1/2"	× 11"

Crank pins:

Outside	5 1/2" × 6"
Inside	8 1/4" × 5 3/4"

Coupling pins:

Leading	4" × 4 3/8"
Driving	6" × 4 1/4"
Trailing	4" × 4 1/2"

Springs. — Bogie wheels: Helical, 10" long free, 5 1/16" outs. diam., Timmis section. — Coupled wheels: Laminated, 11 plates 5" wide × 5/8" thick, 3' 6" crs. — Trailing wheels: Laminated, 11 plates, 5" wide × 5/8" thick, 4' 6" crs. — Bissel: Helical, outs. diam. 5 1/2" × 10 3/16", long free.

Cylinders:

Number	3
Diameter	20"
Stroke	26"

Motion:

Type: Outside	Walschaerts.
— Inside	Gresley.
Valves	Piston.
Diam. of valves	8"
Max. travel of valves	5 3/4"
Steam lap, outs. cylinders	1 5/8"
Steam lap, ins. cylinders	1 11/16"
Exhaust lap	Nil.
Cut-off in full gear	65 %

Tractive effort at 85 % boiler pressure	41 437 lb.
Total adhesive weight	147 840 lb.
Adhesive weight / Tractive effort	3.57
Brake	Vacuum.

pop safety valves are fitted, with a working pressure of 250 lb. per square inch. The steam collector is a steel pressing integral with the dome, the steam supply being taken through a series of 1/2" slots cut in the top of the barrel plate. A regulator of the double beat type feeds a 43-element Robinson superheater through a 7" diameter main steam pipe. The elements are of the short loop type and extend to within 9" of the copper tubeplate.

The boiler is fitted with two injectors, that on the right hand side being a Davies and Metcalfe No. 12 exhaust steam injector, and that on the left hand side a Gresham and Craven 11-mm. live steam under footplate injector.

The three cylinders are each 20" diameter × 26" stroke and are cast separately, the exhaust from the outside cylinders being carried to the blast pipe base through passages in the cast steel saddle. The inside admission piston valves are of the narrow ring type and are 8" in diameter. Separate steam pipes 5" in diameter are fitted to each cylinder.

A « Kylchap » double blast pipe and chimney is fitted having 5 1/4" diameter blast pipe tops. The connecting and coupling rods are of nickel-chrome steel, similar to those fitted to the Company's « Pacific » type engines. The whole of the revolving and 40 % of the reciprocating masses are balanced, the revolving masses at the centre crank being balanced by extensions of the crank webs.

The valve gear arrangement follows the Company's standard practice in which the outside valves are operated by means of Walschaerts gear and the inside valves by the Gresley gear. The maximum cut-off is 65 %, at which position the valve travel is 5 3/4".

The coupled axleboxes are lubricated by means of a Wakefield six-feed mechanical lubricator and Armstrong oiler pads are used in the axlebox trays. A Wakefield mechanical lubricator is also used for the lubrication of the valves and cylinders, one feed being placed in each steam pipe and one to the top of each cylinder barrel. The two lubricators are driven from one return crank on the right hand trailing crank pin. Four oil boxes each with nine syphon feeds lubricate the valve spindles, piston rods, etc.

The locomotive is fitted with vacuum brake. The two 27" diameter brake cylinders are arranged to transmit their load to one main shaft and give a total brake power equal to 93 % of the adhesive weight.

Reversing gear is of the vertical type fitted with ball thrust washers and a vacuum locking device is provided.

Bowden wire is used for the sand gear and cylinder cock controls, as well as for operating the whistle, the latter being fitted immediately in front of the chimney.

Bucket seats are provided for the driver and fireman and flexible rubber roofing is fitted over the gap between the cab and tender.

In addition to the usual cab fittings, a Flaman speed indicator and recorder is fitted.

A turbo-generator is also provided and situated at the front end of the engine. This supplies the engine headlamps, also a number of lamps placed under the skirting and between the frames in order to facilitate engine examination. Lamps are fitted in the cab to illiminate the water and pressure gauges and the speed indicator. When running at night the coupled wheels and motion can be flood-lit.

All wiring is in steel conduits and the switch box is located under the driver's seat.

The tender is of the eight-wheeled corridor type fitted with Pullman vestibule and buckeye coupler, and is equipped with a water scoop. It carries 5 000 gallons of water and 8 tons of coal.

The engine and tender are completely

reamlined, and painted in garter blue nilar to the « Dominion » class engines orking the Coronation train.

The following is a brief comparison between the 4-6-2 A.4 type engines and engine No. 10 000 :

	A. 4 type.	*10 000.*
oiler pressure	250 lb./sq. in.	250 lb./sq. in.
otal evaporative heating surface	2 576.3 sq. ft.	2 597.6 sq. ft.
uperheater :		
Number of elements	43	43
Heating surface	748.9 sq. ft.	748.9 sq. ft.
otal heating surface	3 325.2 sq. ft.	3 346.5 sq. ft.
rate area	41.25 sq. ft.	50 sq. ft.
ylinders (3)	18 1/2" × 26"	20" × 26"
active effort at 85 % boiler pressure	35 455 lb.	41 437 lb.
otal weight :		
Engine	102 t. 19 cwt.	107 t. 17 cwt.
Tender	64 t. 3 cwt.	64 t. 3 cwt.

MISCELLANEOUS INFORMATION.

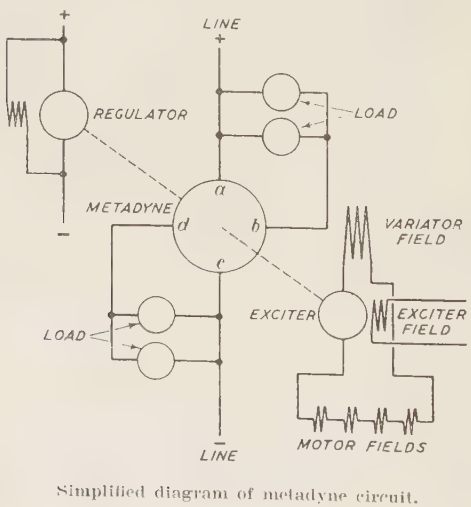
[621. 557]

Electric train control. — The metadyne system.

(*Modern Transport.*)

During the past three years a considerable amount of experimental work has been carried out with a view to the reduction of power losses in the resistances of low-tension direct-current electric trains on suburban service, where frequent stops and restarts have to be made. The work has been undertaken by Metropolitan-Vickers Electrical Co., Limited, and experience of train operation has been gained upon the tube railways and shallow underground lines of the London Passenger Transport Board. These culminated in an order for 58 two-car sets, embodying « metadyne » equipment, the first of which are now in operation and were described in *Modern Transport* of September 4. This makes appropriate further reference to, and explanation of, the metadyne equipment.

The name « metadyne » has been given to a special type of D.C. machine in which special use is made of armature reaction to control the machine characteristic. The special



Simplified diagram of metadyne circuit.

constructions and their applications are covered by numerous patents taken out by Mr J. M. Pestarini, the well-known Italian engineer. The British rights of these patents are held by the Metropolitan-Vickers Electrical Company, which has developed the principle for various applications, including power control on D.C. traction equipments. A notable example of this application is found in the multiple-unit train equipment now being supplied by the Metropolitan-Vickers Electrical Company to the London Passenger Transport Board.

Acting as a transformer.

Metadynes can be built to act as motors, generators or transformers of D.C. power, and it is this last-named type which is applied to the train control. It is arranged to convert power from constant voltage to variable voltage so as to give any desired speed-torque characteristic to the traction motors from standstill to full speed and from full speed to stop. This control is accomplished inherently by the metadyne, which automatically gives a variable boost to the back e.m.f. of the traction motors in order to balance the line volts, overcome the C.R. drop of the machine, and so produce the desired speed-torque characteristic. The metadyne thus takes the place of the starting resistances, accelerating relays, switches, cam groups, etc., of the equipment commonly used for train control and in addition it provides means for the regenerative control of the machine to a standstill. The power wastage ordinarily involved in resistance losses is thereby avoided, while the smooth control has obvious advantages over the use of stepped resistances.

The shape of the speed-torque characteristic can, if it is so desired, be varied at will by the driver. It is possible, therefore, to ar

ge the equipment to operate automatically a schedule or to give the flexibility of control of the steam locomotive, without peaks and drops in tractive effort which involved with the resistance notching and nsitions of standard methods. The circuit rangement used in these equipments is wn in simplified form in the accompanying gram.

The metadyne circuit.

The metadyne set consists of three commutor type machines mounted on a common aft. The machines are :

(a) A regulator, which starts the set and eps it running at a fixed speed.

(b) The metadyne proper, which feeds the natures of the traction motors.

(c) An exciter, which provides excitation the metadyne « variator winding » and fields of the traction motors.

The regulator is in its simplest form a unt machine, while the exciter is a generawith a separately excited field winding. e metadyne itself consists of an armature ordinary type, with commutator, and a stawhich has in this case four main poles d four brush arms.

A constant voltage supply is fed to the primary » brushes « *a* » and « *c* », and the d is connected between brush « *a* » or « *c* » d the corresponding « secondary » brush » or « *d* ». The back e.m.f. of the machine at the brushes « *a* » and « *c* », produced by armature reaction due to the current in the motor circuits, tends to maintain itself constant, any tendency to increase of current in the motor circuit producing a checking increase of back e.m.f. and vice versa. A stable condition is thus produced, in which the motor load current is maintained constant irrespective of the motor speeds.

Motoring and regeneration.

The variator winding provides means for controlling the motor current by superimposing on the axis of armature reaction a separately excited flux, subtracting from the effect of armature reaction, so that more current is drawn from the line and the current in the motor circuits is increased. By suitable control of the exciter field any desired characteristic can thus be given to the motors.

In regeneration in order to avoid excessive voltage on the metadyne brushes the motor armatures are connected across the brushes a—d and b—c. In this case the motors are acting as generators, but here again, since the back e.m.f. tends to maintain itself constant, the currents in the motor circuits remain substantially constant unless modified by the variator winding. The current delivered to the line gradually diminishes as the train speed falls, until at low speeds and standstill the braking effort is actually maintained by current drawn from the line.

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OFFICIAL INFORMATION

ISSUED BY THE

PERMANENT COMMISSION

OF THE

International Railway Congress Association

Revision of the Rules and Regulations of the Association

At the Plenary Meeting on June 11th, 1937, the Paris Congress approved the proposals of the Permanent Commission of the Association, tending to revise Articles 2 and 4 of the Rules and Regulations.

The object of this revision was to simplify the formalities required when altering the list of countries to which the Association extends, and thus facilitate the admission of new members (see page 1962 of the September 1937 *Bulletin*).

In accordance with the Rules and Regulations (art. 22), a written ballot among the affiliated Railway Administrations, Organisations and Governments took place within the period laid down; the result of this ballot is quite affirmative, the votes received in favour of the proposal numbering 832 on a total of 1107 valid votes, no negative vote having been sent in.

The revised Rules and Regulations consequently came into force on December 12th, 1937, i.e. six months after the closure of the Paris Congress, and are worded as follows :

Rules and Regulations of the International Railway Congress Association.

DECEMBER 1937.

Objects, headquarters and constitution of the Association.

ARTICLE 1. — The object of the International Railway Congress Association is to facilitate the progress and development of railways by the holding of periodical Congresses and by means of publications.

The headquarters of the Association are at Brussels.

ART. 2. — The Association is composed of State Railway Administrations, and of Administrations directly concerned with the working of railways of public utility in countries which have joined the Association.

It is composed of the Railway Administrations included in the list which has been published in the *Bulletin of the Railway Congress* for September 1922, p. 1274.

The Governments of the countries to which the Association extends may also join the Association.

The list of these countries (see appendix) may be altered by a decision of the Permanent Commission, as stated in article 4.

International Organisations which deal essentially with railway operation may be admitted as adherents of the Association, and may also National Unions formed of administrations operating at least 5 000 kilometres (3 100 miles) of railways not belonging to the Association.

Permanent Commission.

ART. 3. — The Association acts through a Permanent Commission composed of members nominated according to article 6.

ART. 4. — The Permanent Commission can, by a postal vote, giving a majority of three quarters of the total membership, alter the list of countries belonging to the Association.

It has power to arrange for the admission of Organisations in a position to belong to the Association and of Railway Administrations, subject to a written report prepared by a member nominated in each case by the President.

The only Railway Administrations who may be admitted to the Association, beyond those who constituted it in 1922, are those whose principal business is the running of railways worked by mechanical traction, whether owned or leased by a public authority, open for public traffic and having a length of at least 100 kilometres (62 miles) open for traffic and average receipts per annum, over the last three years of at least two million gold francs (1).

Administrations cease to belong to the Association if they fall within either of the following categories :

a) Undertakings which enter into liquidation or default and Railway Administrations which cease to operate their lines directly in the public service;

b) Members who have not paid their subscriptions for more than two years and who, having been duly notified, have not paid the arrears;

c) Railway Administrations admitted subsequently to the formation of the Association in 1922, which no longer fulfil the conditions laid down in the 3rd paragraph of this article.

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(1) In calculating the length of line, sections of mountain railway worked by special methods of haulage are taken as equivalent to twice their length.

The admission of an Administration in a State which itself is only provisionally included in the Association, under the terms of the 1st paragraph of this article, remains provisional until the admission of the State itself is confirmed.

ART. 5. — The Commission shall organise the Congress meetings, prepare an agenda for discussion, be responsible for their preliminary treatment, edit and publish reports for the discussions, draw up a balance sheet, fix, subject to article 17, the amount of the annual subscriptions, be responsible for the finances, and set on foot any investigations and issue any reports or other publications which will, in its judgment, further the objects of the Association.

ART. 6. — The Permanent Commission shall be composed of ex-presidents of the sessions of the Congress, ex-officio members, and elected members, the number of which shall be fixed by each Congress.

The elected members shall be, as far as possible, chosen to represent the different Nations. Under no circumstances shall any one Nation have more than nine elected members.

One third shall retire at each Congress and shall be eligible for re-election.

Members who have ceased to hold office of any kind, so that they are no longer attached to the railway service of an adherent Government, to an adherent Organisation or to a participating Administration which qualified them at the time of their election, shall cease to be members of the Permanent Commission. It shall be the duty of such a member at once to inform the president when he retires. The Permanent Commission may, upon the application of five members, with the consent of the majority of all its members consulted by letter, ask him to remain until the next Congress.

The Commission shall always have the power to complete its number by co-opting delegates of the participating Administrations and adherent Governments. In this case a final election shall be made at the following Congress.

When the place of meeting of a Congre has been fixed, the Permanent Commissio shall have power to co-opt as temporar members representatives of the country i which the Congress is to be held.

ART. 7. — The Permanent Commission sha elect from among its members a presider and two vice-presidents at its first meetin after each Congress of the Association.

The president and one of the vice-presi dents shall be chosen from the Belgian mem bers.

The Commission shall appoint a genera secretary, a secretary-treasurer and secre taries. As such they shall have the righ to attend the meetings in a consultative ca pacity.

The Commission shall be summoned b the President when the business of the Asso ciation requires it, but in any case at leas once a year.

A meeting may be held when demande by any five members.

Questions shall be decided by the vote of the majority of members present. the votes are equal, the Chairman shall hav the casting vote.

Minutes shall be kept of the proceeding of the Commission. Nine members shall for a quorum.

If at a first meeting this number is no present, the meeting shall be adjourned fo not longer than one month, when it sha then be considered valid whatever may b the number of members present.

ART. 8. — The Association shall be re presented in Courts of Justice and in a civil actions by the president, or in the even of his being unable to attend, by the genera secretary of the Permanent Commission.

Executive Committee.

ART. 9. — At its first meeting after a Con gress the Permanent Commission shall no minate three of its members, who, with th president and the vice-presidents of th Commission, shall form an Executive Com mittee.

The president of the Permanent Commission shall also be president of the Executive Committee.

The general secretary, the secretary-treasurer and the secretaries of the Commission shall be members of the Committee, and hall have the right to take part in discussions, but not to vote.

The members of the Executive Committee hall hold office for a period equal to the nterval between two Congresses. They shall e eligible for re-election.

The Executive Committee shall meet at he instigation of the president on his own uthority or at the request of three members.

The Committee shall be responsible for he management of current affairs and for inancial business, also for superintending nd managing investigations, reports and publications; for the editing of the *Bulletin* and or the care of the library and archives. It shall ecide as to printing in whole or in part ne reports and other documents submitted o a Congress, the circulation of which it egards as necessary to facilitate the discussions. It shall be the duty of the Committee to furnish members of the Association with such special information at they ay require.

The Committee shall have the right to ppoint and to dismiss the staff.

The carrying out of the decisions of the ommittee is entrusted to the president and ne general secretary.

Congresses.

ART. 10. — At each Congress the Association shall fix the time and place for the llowing Congress.

The Permanent Commission may alter such rrangements under exceptional circumances.

ART. 11. — The following shall have a right attend the meetings :

1) The members of the Permanent Commission, the secretaries and the secretary-easurer;

2) Delegates appointed by the participating administrations and adherent Governments and Organisations.

These must belong to the regular and permanent staff of the delegating Administration;

3) The secretaries of sections, and the reporters appointed by the Permanent Commission or by the Executive Committee to report on the items on the agenda.

ART. 12. — The participating Railway Administrations shall be entitled to nominate a number of delegates in proportion to the mileage of their system in the following ratio :

One delegate for lines of not more than 200 km. (124 miles);

Two delegates for lines of 200 to 500 km. (124 to 311 miles);

One additional delegate for each additional 500 km. (311 miles) or fraction thereof above 500 up to 3 000 km. (1 864 miles);

One additional delegate for each additional 1 000 km. (621 miles) or fraction thereof above 3 000 up to 6 000 km. (1 864 up to 3 728 miles);

One additional delegate per 2 000 km. (1 242 miles) of fraction thereof above 6 000 up to 10 000 km. (3 728 up to 6 214 miles);

One additional delegate per 4 000 km. (1 242 miles) or fraction thereof above 6 000 km. (6 214 miles).

The adherent Governments and Organisations may nominate delegates to a maximum number of 10 in the proportion of one delegate for each 100 gold francs subscription up to 500 gold francs, and one additional delegate for each 250 gold francs over 500 gold francs.

The maximum figures quoted above do not include those members who take part in the Congresses in accordance with the 1st and 3rd paragraphs of article 11, and whom the participating Administrations, or adherent Governments and Organisations include amongst their delegates.

All delegates participate in the Congresses with equal rights.

ART. 13. — At the opening of each Congress the officers of the Permanent Commission shall hold office temporarily, and

the Congress immediately elect its own officers as follows :

1) One or more honorary presidents and vice-presidents;

2) One president;

3) One or more general secretaries and one or more assistant secretaries.

The presidents of sections, elected as explained in article 14 following, shall also be included.

The delegate nominated first by each government shall be an ex-officio vice-president.

All officers shall be appointed for the duration of the Session.

The election shall take place according to the rules laid down in article 16, paragraph 6. The duties of the officers shall be those laid down by standing orders adopted by deliberative assemblies for the conduct of their proceedings.

After the appointment of the officers, the Congress shall resolve itself into sections, according to the arrangements of items on the agenda submitted by the Permanent Commission.

A member may enter his name for more than one section.

The Congress may also appoint special committees to investigate certain questions.

ART. 14. — Each section or committee shall appoint its own officers consisting of :

1) A president;

2) One or more vice-presidents;

3) One or more principal secretaries and secretaries.

The principal secretaries of each section or committee, however, shall be nominated by the Permanent Commission.

The sections and the special committees shall cease with the termination of each session.

ART. 15. — The discussions of the Congress shall be confined to the questions set down in the agenda for the session by the Permanent Commission. The latter may receive suggestions, either as regards the inclusion of a question or relating to a particular question already raised by participants or adherents.

A reporter appointed by the Permanent Commission, shall prepare a brief outline of each question placed on the agenda, together with a resume of the documents he has been furnished with; he will not draw up the final summary.

No question shall be discussed at a general meeting without having been first considered by a section or special committee.

ART. 16. — The discussions shall be conducted in French and in the language of the country in which the Congress is held. Speeches in any other language shall be translated into French.

The minutes and reports shall be drawn up in French, but speakers on demand shall be entitled to have their original words reproduced.

The officials of the respective sections shall draw up an abstract of the discussions setting forth the various opinions expressed in the section. After receiving the approval of the section these abstracts shall be submitted to the general meeting. They shall then be inserted in the minutes after having been completed by the addition, if necessary, of any new opinions expressed at the general meeting itself.

The Congress shall not vote except on questions of management or organisation.

On these special questions, the votes of the majority of the members present shall be taken by members rising in their places. If there is any doubt the votes shall be counted. A roll call shall not be held except at the request of not less than twelve members.

Subscriptions and auditing of accounts.

ART. 17. — The expenses of the Congresses of the Permanent Commission and of the Executive Committee shall be covered by

1) The annual subscriptions of members;

2) Subsidies and other casual receipts.

The annual subscription shall consist of

a) In the case of adherent Governments

nd Organisations, an amount fixed by themelves, which in the case of adherent Organistions shall not be less than 100 gold francs;

b) For the participating Administrations, fixed sum of 100 gold francs plus a sum roportionate to the length of the system. his variable sum is fixed by the Permanent ommission, but may not exceed the fifth f a gold franc per kilometre.

The financial year shall begin on the st January.

ART. 18. — Payment of the subscriptions ntitles participating Administrations, Goernments and adherent Organisations to eceive free as many copies of reports, of he proceedings and other publications as he number of their delegates.

ART. 19. — The Permanent Commission hall present to each Congress a report on he financial position. The Congress shall ppoint two auditors to pass the accounts.

Revision of the constitution, dissolution and liquidation.

ART. 20. — The rules may be revised by he Congress on the proposal of the Permaent Commission, due notice being given to he participants and adherents by a letter ent out at least two months before the pening of the session.

Proposals for modifications put forward by articipants or adherents must reach the Permanent Commission at least six months beore the opening of the session. If they are dopted by the Commission, they are submitted to the Congress by means of a report ent out by the Commission, which should e despatched to the participants and adheents at least two months before the opening f the session.

If the Permanent Commission does not ccept a proposed modification, it shall not e submitted to the Congress, unless it is upported in writing by participating Admiistrations or adherent Governments or Organisations entitled to be represented by 00 delegates at the Congress. In this ase the proposition is submitted to the participants and adherents by a letter from the Permanent Commission sent out at least two months before the opening of the session.

ART. 21. — Every proposal for amending the rules shall be discussed by the Congress on a report of a special committee presided over by the President of the Permanent Commission, or by his delegate and composed of eleven members, as follows :

a) The President of the Permanent Commission or his delegate;

b) Four members delegated by the Permanent Commission;

c) One delegate from each of the five sections which form the Congress, this delegate being chosen after discussion of the suggestion by the section;

d) One delegate representing the authors of the suggestion.

ART. 22. — The proposed modification shall only be considered if it is supported by the clear majority of the delegates present at the general meeting.

The modification shall not be finally adopted until a written ballot has been taken, which shall be called for by a circular letter sent out by the Permanent Commission during the month following the termination of the session of the Congress. In this ballot each Administration, Government or Organisation belonging to the Association shall have a number of votes equal to the number of the delegates to which it is entitled.

The ballot shall be closed six months after the termination of the session of the Congress.

Only those modifications will be adopted which are supported by two-thirds of the votes received at headquarters of the Permanent Commission by the end of this period.

The results of the ballot will be published in the monthly *Bulletin* of the Association.

ART. 23. — Adherent Governments and Organisations, also participating Administrations constituting the Association pledge themselves to promote the meetings of the

Congress and the work of the Permanent Commission.

ART. 24. — The Association may be dissolved only by a three quarters majority, on a postal vote from the participating Administrations, each one having a number of votes equal to that of the delegates to which it is entitled.

ART. 25. — In the event of dissolution, the Permanent Commission shall have authority to arrange the liquidation.

The assets of the Association are to be distributed by its agency among works conforming with the objects of the Association or among philanthropic works in connection with railways.

ART. 26. — In no circumstances shall participating Administrations and adherent Governments and Organisations who for any reason have ceased to belong to the Association, have any claim on the assets of the Association.

* * *

APPENDIX.

LIST OF COUNTRIES

included in the International Railway Congress Association.

Argentine;
Austria;
Belgium and Colony;
Bolivia;
Brazil;
Bulgaria;
Chili;
China;
Colombia;
Costa Rica;
Cuba;
Czechoslovakia;
Denmark;
Dominica;
Ecuador;
Egypt;
Finland;
France, Algeria, Tunisia, Colonies and Protectorates;
Germany;
Great Britain and North of Ireland, India, Dominions, Protectorates and Colonies;
Greece;
Hayti;
Holland and Colonies;
Hungary;
Irish Free State;
Italy;
Japan;
Jugoslavia;
Luxemburg;
Mexico;
Nicaragua;
Norway;
Paraguay;
Peru;
Poland;
Portugal and Colonies;
Rumania;
Salvador;
Siam;
Spain;
Sweden;
Switzerland;
Turkey;
United States of America;
Uruguay.